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AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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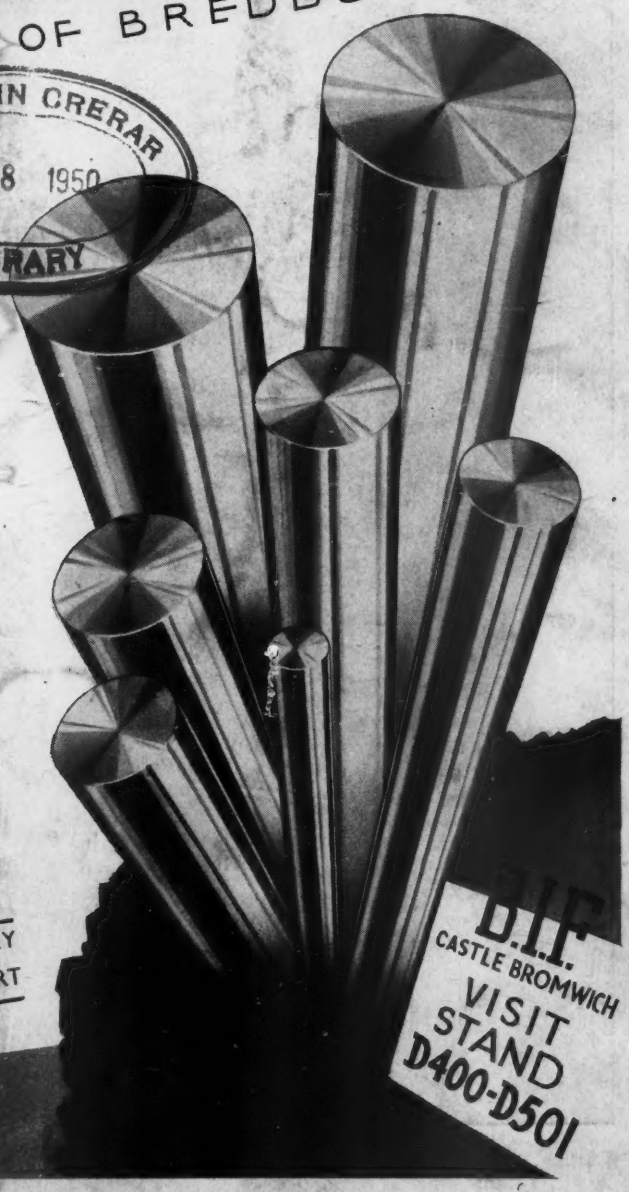

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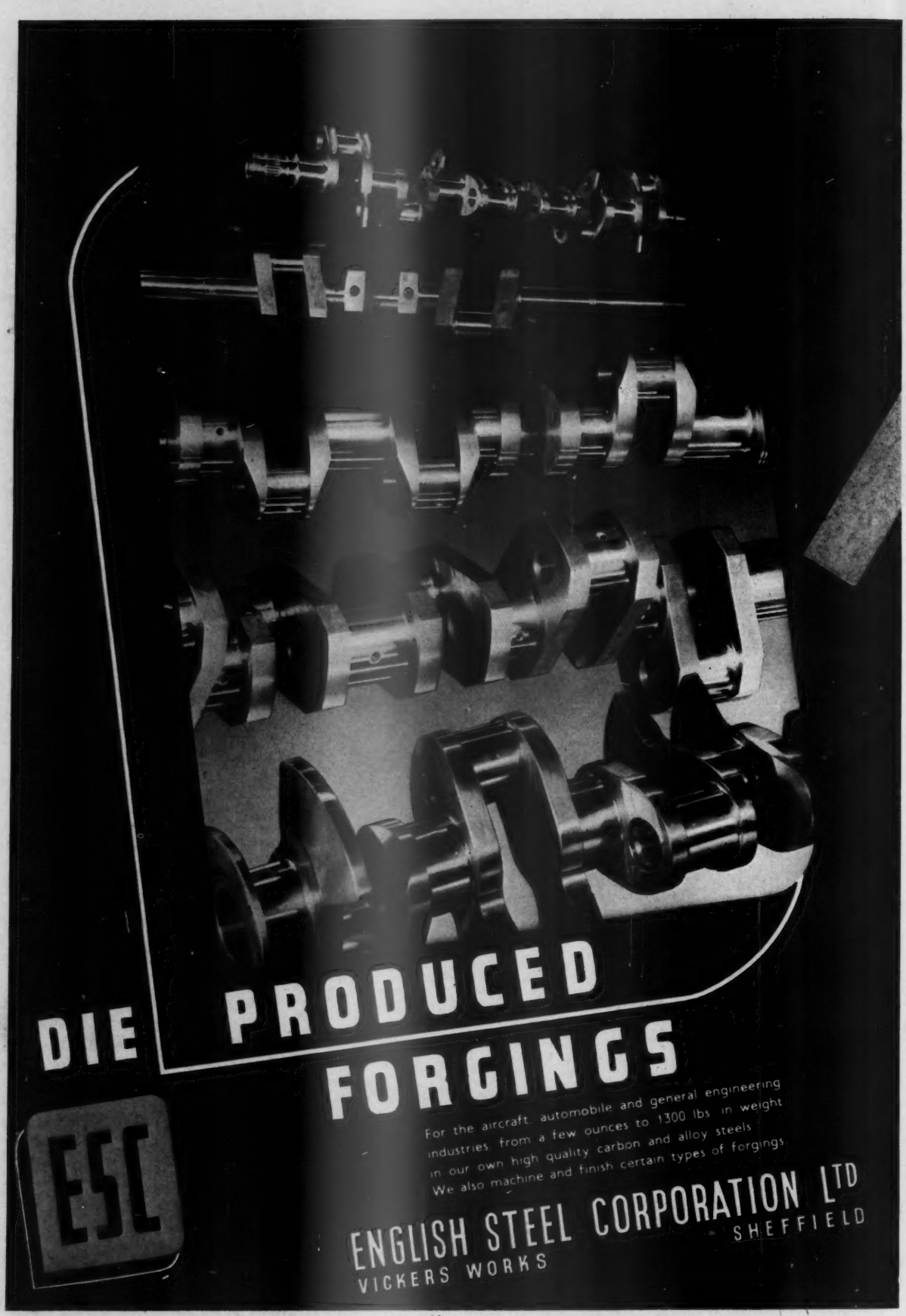
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Design, Materials, Production Methods, and Works Equipment

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The Gas Turbine

SOONER than was generally anticipated, a car driven by a gas turbine has been publicly demonstrated. Whatever may ultimately emerge from the event, this occasion will certainly prove to have been the most significant happening since the beginning of automobile history. There can be little doubt that it foreshadows the most radical departure in the technical conception of the road vehicle so far envisaged.

A wise decision

The Rover Company, to whom the major share of the credit for this innovation is due, have stated with commendable restraint that "we know we still have a long way to go before the gas turbine car becomes a marketable proposition." Nevertheless, their decision to demonstrate the vehicle at this early stage of development was most certainly a wise one.

It is fairly certain that similar projects are in hand elsewhere, and in times when international commercial rivalry is so intense, publicity of this kind has always been employed as a major weapon of offence. Moreover, as such publicity is forced upon the individual's attention to the discouragement of discriminating thought, it is important that the lead so long held by Great Britain in scientific development and inventiveness should be emphasized by deed as well as word.

A momentous event

It was inevitable that for many of those present at the demonstration, the occasion in itself tended to overshadow the vehicle. However, probably few were prepared for quite so finished a performance. A comparatively minor point that also merits commendation was the way in which the vehicle itself was presented. Notwithstanding that, in the words of the Rover Company, the vehicle is a piece of test bed apparatus, it might well have found a place on any exhibition stand.

Since the gas turbine became a practicable prime mover in large sizes, its application to road vehicles has been the subject of numerous abstract examinations. In nearly every case the authors' conclusions might be briefly but fairly summarised as "guarded pessimism." Indeed it is remarkable that the possibility of so radical a change in automobile design has not only provoked relatively little discussion but has attracted no forceful protagonist from the ranks of engineers. Naturally those most intimately concerned with the concrete attack on the problem could not be expected to disclose their progress and intentions. It is, however, a curious fact that no authoritative voice has so far been raised in the cause of the turbine as a power unit for road transport vehicles.

The unoptimistic view

Whatever may be the reason, the impression remains that in general the tendency has been to emphasize the disabilities of the turbine, real and imaginary. In fact, a certain short-sightedness can be detected in this concentration on the practical difficulties in these early stages. All things considered, a more whole-hearted faith in the ultimate success of the engine might have been anticipated. From this point of view, the recent demonstration must result for many in a revised outlook. Although the problems yet to be solved remain considerable, the final outcome can no longer be seriously doubted.

The bare facts of the observed performance are recorded in the Royal Automobile Club's test report. The time taken to start the power unit and to run up to idling speed was 13·2 seconds and the car moved forward in a further 3·4 seconds. The car was driven for five laps on a circuit of the Motor Industry Research Association Proving Ground, measuring approximately 2·75 miles per lap. No attempt was made to attain maximum speed, but during the course of the test a speed exceeding 85 miles per hour was readily attained. At this speed the compressor-turbine revolution counter indicated 35,000 r.p.m. In a test of acceleration from standstill the car smoothly attained 60 miles an hour in 14 seconds.

Taken alone the figures do not necessarily establish a fully convincing case, as many who are experienced in conducting and interpreting the results of road tests will

agree. Figures in themselves, however, are but a part of the whole that makes an acceptable vehicle. To observe the vehicle in action, when ease of control, manœuvrability, noise and so on, can to some extent be assessed, leaves little doubt that the promise of the turbine is likely to be fulfilled. It is evident that even now, with the development of a suitable heat exchanger, a turbine engined car would be within measurable distance of challenging the larger American vehicles in performance, having due regard also to fuel consumption.

American design policy

While acknowledging to the full the skill and enterprise of American engineers that has resulted in a fully automatic transmission being offered at a remarkably small premium of $158 on one of the lowest priced American cars, the price that must be paid in efficiency and complication of the mechanism is a heavy one. Even allowing for an emergency low gear in the turbine transmission, the low weight and mechanical simplicity of the turbine by comparison with a six- or eight-cylinder reciprocating engine with automatic transmission, makes a really startling contrast. Given the same production technique, a complete turbine installation could doubtless be made for less than the cost of an automatic transmission alone, and give two-pedal control.

Further, it is hardly logical to over-emphasize the present relatively low thermal efficiency of the turbine while at the same time the cooling systems of conventional designs are increasing in capacity by perhaps 15 to 20 per cent so that transmission losses can be dissipated. Tank miles per gallon is the user's standard of comparison. On this basis it appears that the turbine could already approach the customary economy, if that is the appropriate word, offered by a 4- to 5-litre car with fully automatic transmission.

Making due allowance then for the great amount of development still necessary before a reliable and serviceable turbine unit could be offered to the public, the time seems already in sight when it may be expected to present serious competition to the large car of conventional design.

The reciprocating engine

Doubtless the reciprocating engine can be developed very much further in economy, but this may do little more than offset the increasing transmission losses as simpler and simpler control is demanded. Taking the long view there seems no question as to what course automobile development should take, and to those who hold progressive opinions the Rover achievement will come as encouragement and a stimulus.

At one time when the advent of a small turbine of practicable size seemed at least a possibility, there was a good deal of speculation as to what form a satisfactory transmission might take. With revolutions soaring into fabulous regions, this point was regarded in some technical circles as presenting a major problem. In the final upshot, however, the inherent characteristics of the turbine power unit as such, provided its own answer to the problem. It embodies in fact, in its own working cycle a ready made torque converter. In conveying in broad general outline this aspect of the workings of the device, probably the best analogy would be to describe its working as that of a fluid torque converter, but using a gas instead of a liquid. Further, there has been some misconception regarding the actual speed of the power turbine as distinct from the compressor. While no definite figures are forthcoming, there seems little doubt, from the nature of the gearing employed, that the figure is lower than is still popularly supposed.

Vehicle size

Some considerable time back when the ultimate emergence of a small turbine was first known to be not too far distant, it was confidently assumed that it would first be put forward as a power unit for a heavy vehicle. It was not thought to be likely that such a scheme could be developed in a size small enough for installation in a vehicle of the dimensions of a private car. In the initial stages at any event, it was thought that the smallest possible unit would be much too large and powerful for such a purpose. Here again, general prognostication has been falsified and the turbine makes its first appearance in a vehicle of private car size, and of modest dimensions at that.

A great achievement

Every congratulation then to the Directors of the Rover Company and their able staff, and to the many scientists whose co-operation has brought this notable event about. Not all can be enumerated or even known, but Henry Wiggin & Co. Ltd., who are responsible for the development of the Nimonic alloys used, should be credited with furnishing the material that eventually enabled the designers to overcome one at least of the major difficulties. The work of Joseph Lucas Ltd., in developing the injection equipment much assisted the attainment of a satisfactory working solution to the fuel supply problem.

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THE A.E.C. "REGAL" MARK IV

A New Chassis for Coaches and Single-deck Buses

IN private car design, the engineer has to fit the propulsion machinery into any otherwise unwanted space. Further, the passengers nowadays have to mould their bodies into the shape ordained by the stylists. The designer of a public service vehicle, on the other hand, has maximum space as his first aim in order to accommodate in comfort as many fare-paying passengers as possible. Not only must they be accommodated, but entry and exit must be as safe and straightforward as possible, and the standard of comfort must be high. Further, the road behaviour of vehicles destined to carry the public must be beyond possible reproach, and it is also now accepted that too great a strain must not be placed on the driver if he is to control the vehicle properly, and therefore safely.

Double-deck buses for use as stage carriages have reached a highly developed state of design, fully satisfactory in service in closely built-up areas. The lower half of a double-decker does not, however, yield a really satisfactory single-decker. Considerable efforts have therefore been made by designers, both in this country and abroad, to break away from convention and to produce a vehicle that will be satisfactory to the fare-paying public, the operators, and traffic managers, be of sound engineering conception and also be a reasonable production job.

The latest A.E.C. "Regal" Mk. IV single-deck bus or coach chassis, though entirely novel, embodies a great number of working parts identical with those employed in the makers' "Regal" Mk. III chassis. Servicing and the spare part question are therefore materially assisted. This new A.E.C. chassis will shortly enter the production stage and will be available as a 7ft 6in or 8ft wide, right-hand drive model for the home market, or as right- or left-hand drive, 8ft wide, long wheelbase versions for overseas.

SPECIFICATION

ENGINE : A.E.C. 9.6 litre, horizontal six-cylinder, direct injection diesel with toroidal cavity pistons. 120 x 142 mm. bore and stroke, 125 b.h.p. at 1,800 r.p.m.

TRANSMISSION : Improved design fluid flywheel and four-speed, air-operated, self-change gearbox, giving forward ratios of 1, 1.64, 2.53 and 4.50 to 1. Open propeller shafts with Hardy Spicer couplings.

REAR AXLE : Fully floating, underslung worm drive, offset. 8in centres. Ratio 4⅞, 4⅝, 5⅛ or 5.8 to 1.

SUSPENSION : High deflection leaf springs all round.

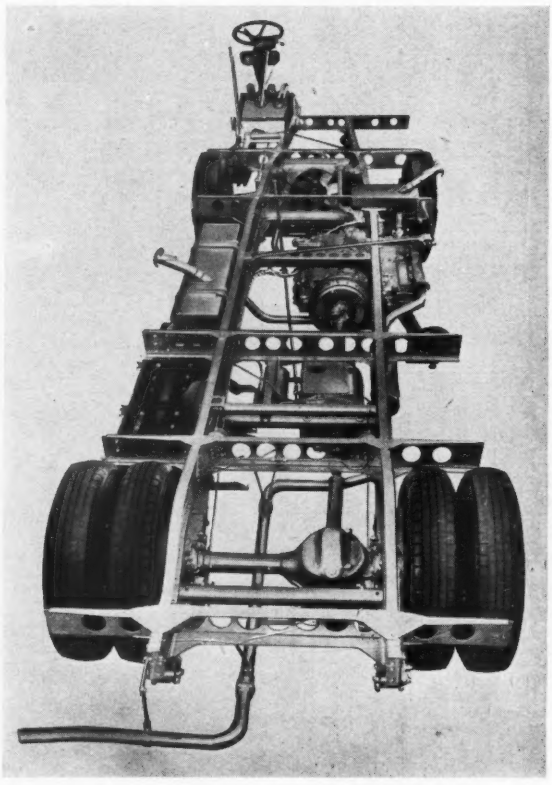
BRAKES : A.E.C. air-operated on four wheels by pedal. Hand brake on rear wheels only.

TYRES : 9.00—20 on 7ft 6in wide chassis, 11.00—20 on 8ft wide models.

DIMENSIONS : 7ft 6in chassis :— 16ft 4in wheelbase. 8ft chassis :— 17ft 6in wheelbase.

As will be seen from the illustrations, this chassis constitutes the body-builders' ideal in having an almost flat and unencumbered platform on which to build the body. It will readily accommodate a single-deck bus-type body seating forty passengers, all facing forwards. This objective has been attained by locating all the components, with the exception of the steering column and driver's controls and a small header tank for the engine coolant, below frame level.

Extending well forward of the front axle the frame carries the steering column and driver's controls at right or left hand as desired. It affords perfect visibility for the driver and what is probably the simplest alternative drive position ever built. The horizontal engine is mounted behind the front axle on the right-hand side of the vehicle under the frame side member. Instant accessibility to valve adjustment or fuel pump, and ready removability of the engine, are outstanding features. The drive is taken through a fluid flywheel to the four-speed pre-selective epicyclic gearbox, thence to an off-set underslung worm drive rear axle. The gearbox and brakes are actuated by compressed air. Valuable experience was gained by A.E.C. from a flat-engined chassis built in 1939 and operated in Canada for experimental purposes during the War.

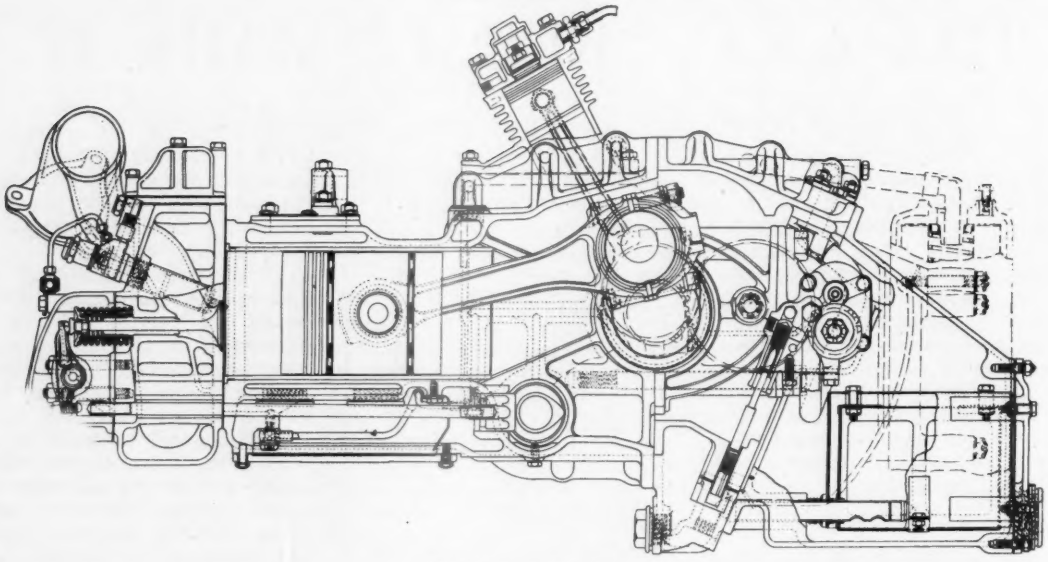


The A.E.C. " Regal " Mark IV.

Engine

Based in design on the Company's conventional 9.6-litre engine, this new horizontal engine also has six cylinders of 120 mm (4·72in) bore and 142 mm (5·59in) stroke. It develops 125 b.h.p. at 1,800 r.p.m. and yields 430 lb/ft torque at 1,000 r.p.m. Compression ratio is 16 to 1 and maximum b.m.e.p. is 110 lb/sq in (7·73 kg per sq cm). Consideration is being given to the possibility of also making available a 130 mm bore version of the same engine to meet arduous overseas operating conditions.

The static portion of the engine is divided into four principal sections, monoblock cylinders and crank-



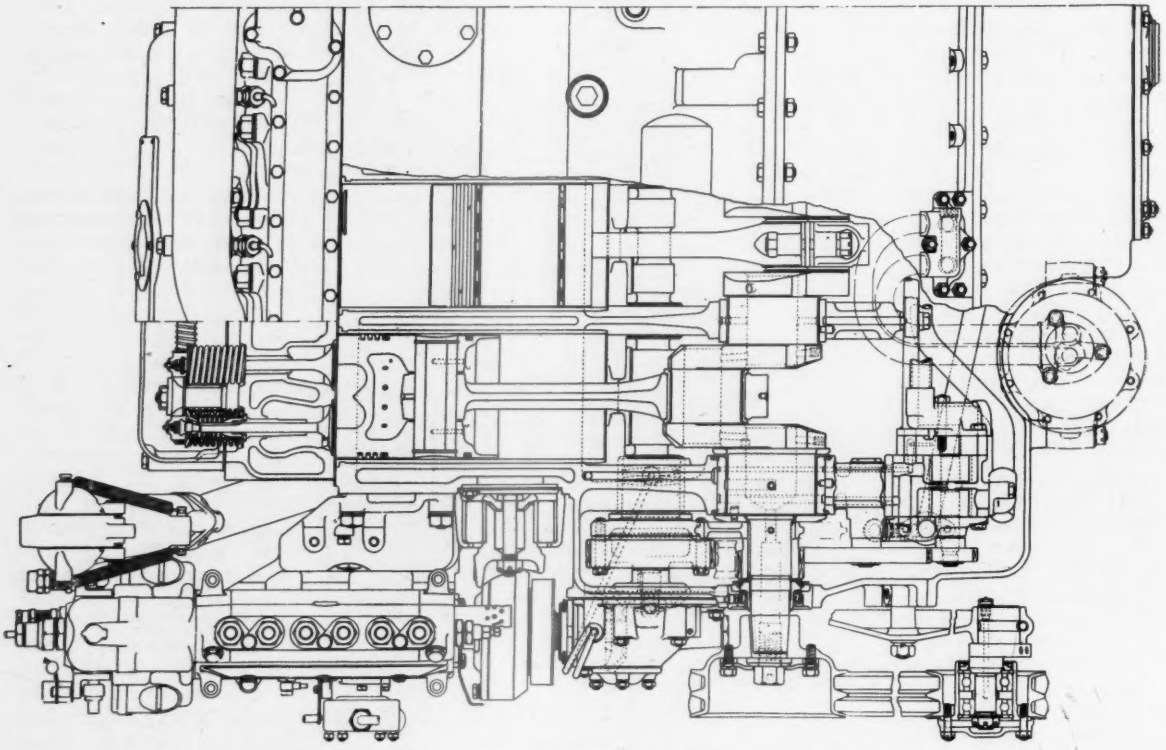
Cross-section of engine.

case, the cylinder head (in two units), the crankcase extension, and the sump. The cylinder portion is an alloy iron casting with renewable centrifugally cast iron liners, a press fit being employed. The crankshaft is carried by seven bearings in this portion of the engine casing which terminates at a vertical line drawn through the centre of the crankshaft.

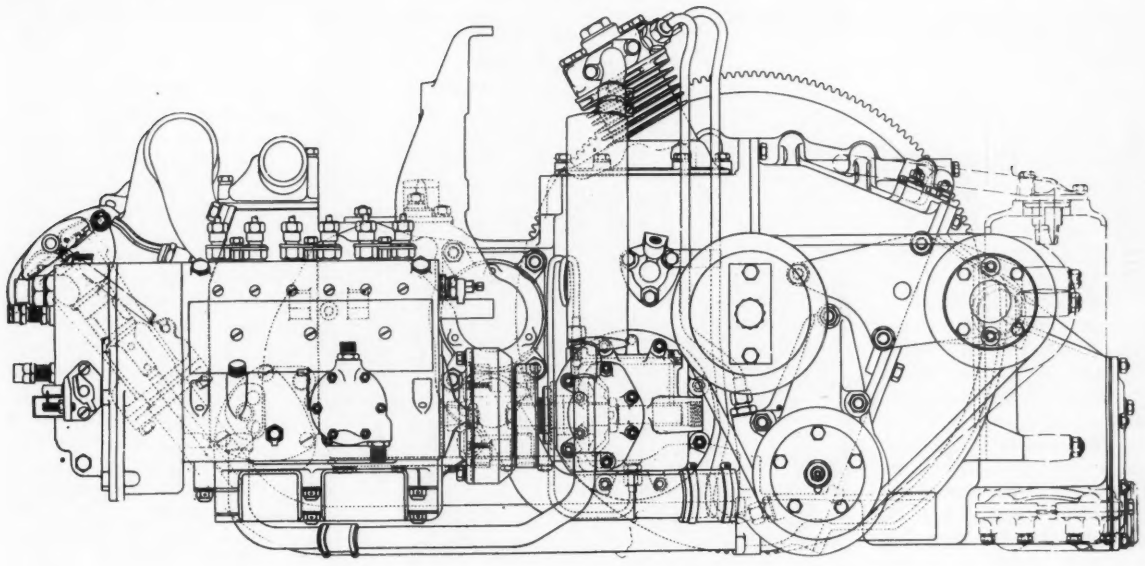
To facilitate handling, the cylinder head is cast in two interchangeable portions. Plates provide access for cleaning the cored waterways. In order to accommodate unevenness in expansion and contraction between the heads, the blackheart cast iron exhaust manifold is also made in two portions.

Both the crankcase extension and the sump are cast in magnesium alloy. Lubrication of a horizontal engine is conveniently effected by the dry sump method, although necessitating two separate oil reservoirs. In the A.E.C. engine, the well to which surplus oil drains is formed in the base of the crankcase extension. The sump is a separate unit having on its inner side a wall for about half its vertical height in order to form a reservoir for $5\frac{1}{2}$ gallons (25 litres) of lubricant.

Forged in alloy steel, the crankshaft



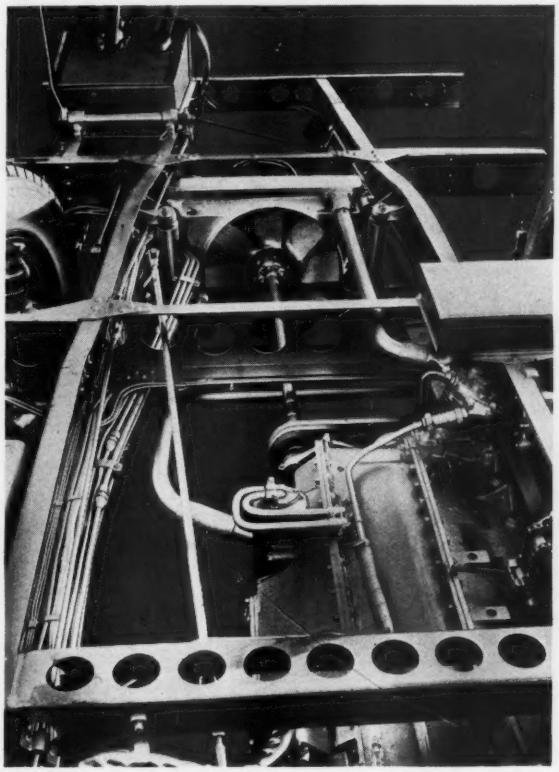
Longitudinal section of engine.



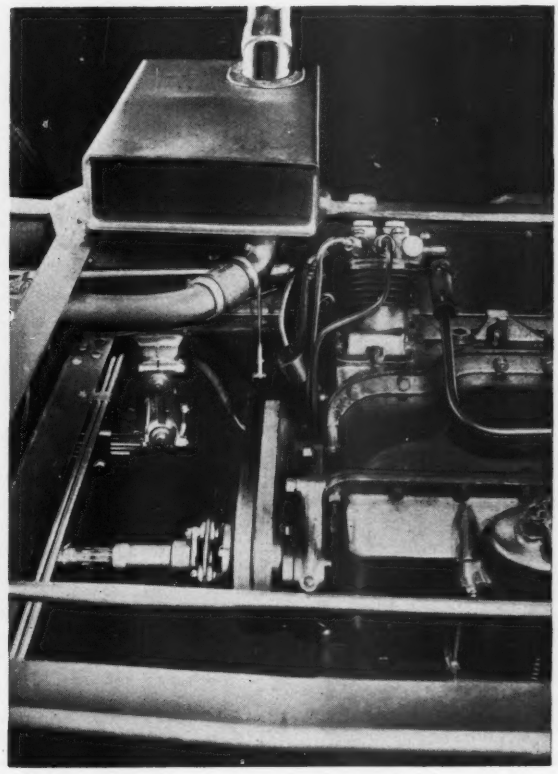
End view showing auxiliary drives.

is of generous dimensions in order to deal with torsional vibration. The main bearings supporting it are of the thick wall type, the "upper" halves being lined with white metal, while the cap halves are lined with lead bronze. All main bearings are 95 mm (3.75in) in diameter. Bearing lengths are:—front 60 mm (2·3622in), centre 70 mm (2·7559in), rear 78 mm (3·0709in) and intermediates 42 mm (1·6535in). The total projected bearing area is therefore 55·33 sq in. Crankshaft thrust is taken by the front main bearing and the crankpins and journals are hollow, and have cross-holes drilled for lubrication purposes. The main bearing caps are each secured by set bolts and located by two dowels. Oil sealing is effected by a synthetic rubber seal at the front end and at the rear end of the crankcase by a series of grooves in combination with an oil thrower formed on the crankshaft.

Of heat-treated aluminium alloy, the pistons are pre-shaped to ensure



Cooling system layout.



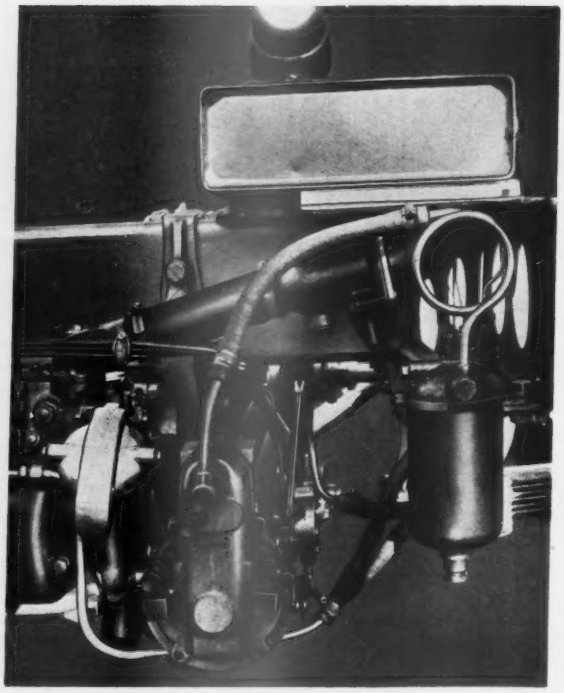
Air compressor and fan drive.

minimum clearances under working conditions. Three pressure rings and one scraper are normally fitted but provision is made for the fitting of an extra scraper ring in each piston skirt as the need arises. The 40 mm diameter gudgeon pins are fully floating and are located by Seeger circlips.

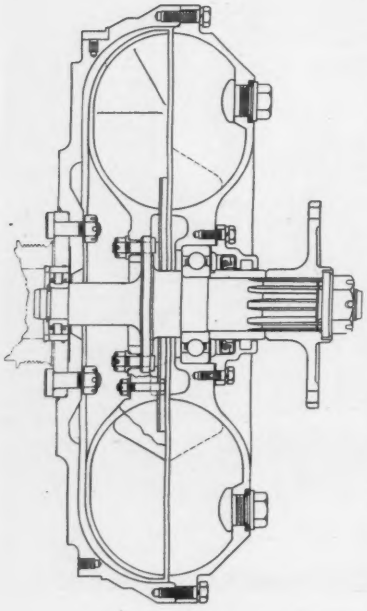
The connecting rods are H-section alloy steel stampings, polished all over, and the solid type small ends are fitted with bronze bushes. Of the waisted type, the centres of the big-end bolts are enlarged in diameter in order positively to locate connecting rod and cap. The heads of the bolts are splined and cut their own way in the rods on fitting. Non-rattle split cotter pins are used in the castellated nuts.

Cast in a special alloy iron the camshaft is carried in seven bronze bushes, thrust being taken by a large collar on the front bearing. The cams directly actuate large diameter piston-type tappets in chilled cast iron. Operated by push rods and rockers, the overhead-valves have push rods made from tube with the ends induction soldered. The rockers have detachable and adjustable hardened ball ends.

A train of robust helical gears constitutes the timing gear. A spring-loaded idler wheel in three sections, the outer portions being circumferentially spring loaded, is provided to take up backlash, also a vernier adjustment to furnish precision setting of the camshaft gear wheel. The fuel injection pump is driven from the timing gear train by a spiral bevel gear.



Fuel filter and injection pump mounting.



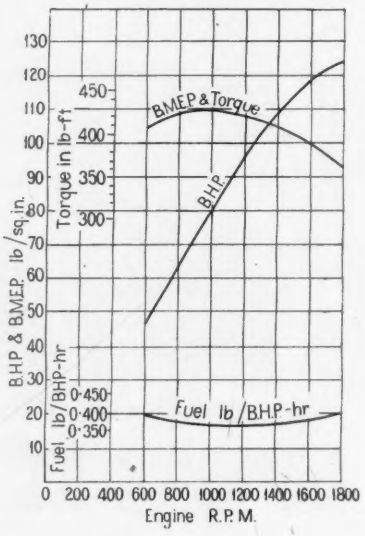
Arrangement of fluid flywheel.

Lubrication of the engine is effected on the dry sump principle, both pressure and scavenge pumps being of the gear type, driven through helical gearing from the front of the crankshaft. The pressure pump draws oil from the rear well of the sump through a gauze strainer which is removable for cleaning purposes. The oil is forced under high pressure to all main and big-end bearings and to the bearings of the idler wheel of the timing gear train. Oil under low pressure is supplied to the front camshaft bearing and the valve gear, surplus oil lubricating the tappets and the remaining camshaft gears. It then drains to a reservoir at the bottom of the crankcase extension, whence it is drawn by the scavenge pump and pumped into the sump.

If specially desired by customers, a felt element type external oil filter can be incorporated in the scavenge circuit. Provision for attachment and connection is made on the upper portion of the sump. The oil filler is at the extreme front of the engine on the off-side, and has a hinged, spring-loaded captive cap. The dipstick is fitted inside the filler neck and has a tapered shank which acts as a seal in order to prevent oil leakage due to a surge of lubricant when the vehicle is cornering. Crankcase pressure is vented through a duct at the rear end which is connected to the inlet air cleaner.

The centrifugal cooling water pump, fitted with a self-adjusting rubber and carbon sealing gland, is built into the bottom of the engine at the front and is driven by twin vee belts from the crankshaft pulley. These belts also drive a third double pulley from which a shaft drive is taken, through Layrub flexible couplings, to the radiator fan. A by-pass type thermostat is fitted in the cooling circuit.

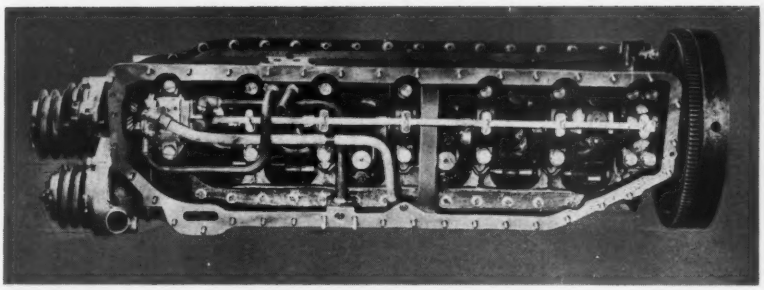
Built integrally into the engine is a single cylinder air compressor which has a free air capacity of 9·87 cu ft (279·5 litres) per minute at 1,800 r.p.m. It is mounted on the wall of the upper end of the crankcase at the front, and has an air-cooled cylinder barrel and a water-cooled head. The cooling water connections are taken directly to and from the water pump casing. The connecting rod which is driven from an eccentric pressed on to the crankshaft is directly white metal lined in the eccentric strap and has a bronze bush in the small end. Driven by the spiral bevel gear from the front of the camshaft the fuel



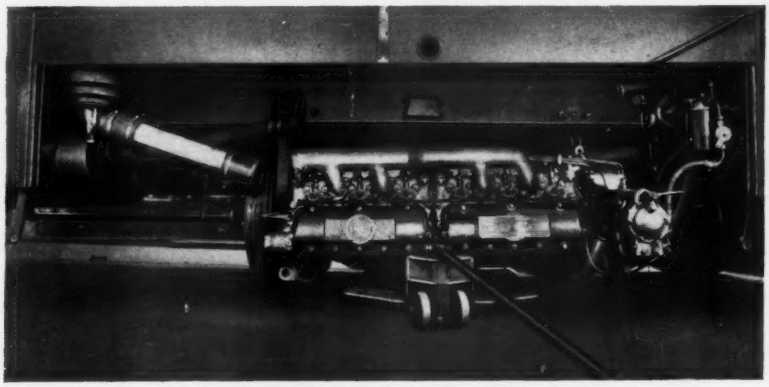
Performance curves.

injection pump is mounted across the front of the engine. The coupling incorporates a flywheel with the object of damping out the pressure impulses. The twin couplings have rubber centres.

The equipment used is the new C.A.V. pilot injection type which, with its two-stage injection, results in smoother idling, better acceleration and considerable reduction of the objectionable "diesel knock." Idling and maximum speeds are controlled by a centrifugal governor which is built on to the back end of the injection pump. Also mounted on the side of the injection pump and driven from the injection pump camshaft is the fuel lift pump. It is of the vertical diaphragm type. Mounted outside the cylinder head covers, each injector embodies a disc-type filter. Accessibility is such that the fuel pump can be removed in 3½ minutes and the injectors in 5 minutes.



Crankcase with cover removed.



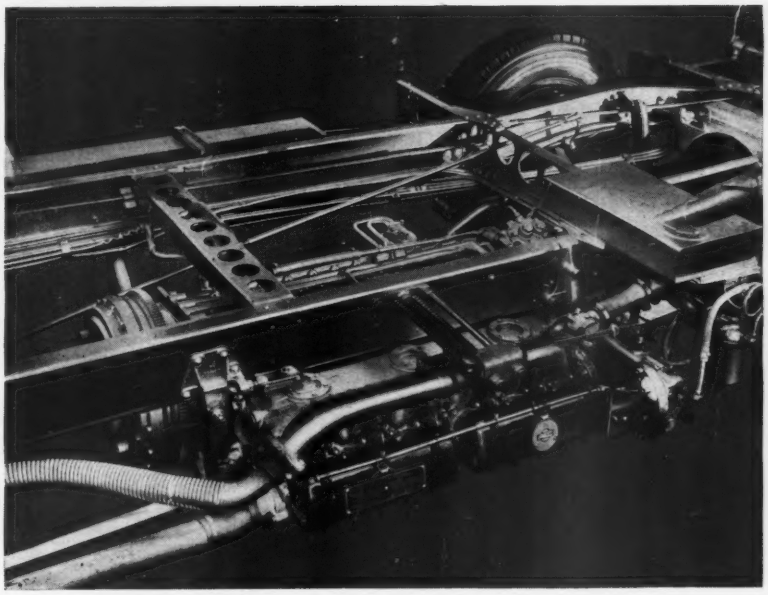
Removing the engine from the chassis.

Air for the engine is drawn through an oil bath air cleaner, which is mounted independently on the right-hand of the frame behind the engine. Connection is made by means of a large diameter flexible pipe.

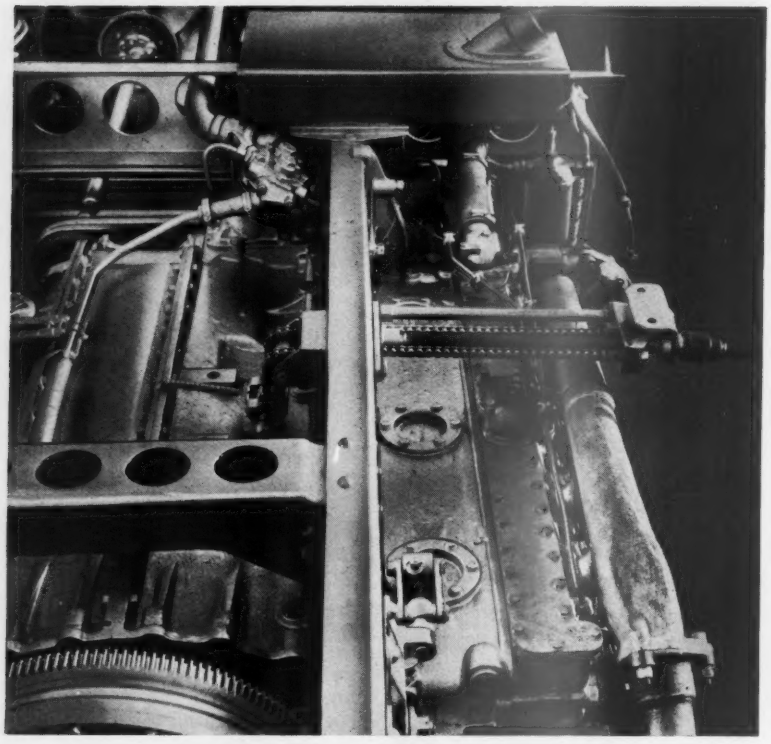
A 24-volt axial type starter motor is mounted on the sump. Its steel pinion meshes with a detachable hardened steel gear ring fitted to the fluid flywheel casing, the ratio being 13:158.

The engine mounting is distinctly novel and the units have been developed by the Metalastik Company in association with A.E.C. It practically amounts to a two point suspension with controlled movement. The front engine support bracket fits around the outside of the right-hand frame side member in line with the air compressor. At the bottom, this bracket supports the front of the engine by a large 2in internal diameter bonded rubber and metal bush. Immediately under the back of the engine is a downswept tubular cross member. This member is bolted to a rigid bracket on the left-hand frame side member and is supported at the right-hand side by a swinging link, rubber bushed at both ends. The cross tube carries a vee-shaped mounting of bonded rubber and metal, to which the rear end of the engine is fixed. The mounting is so disposed that the rubber is in shear to counteract torsional oscillations. Engine torque reactions are absorbed by rubber buffers and a rubber bushed link mechanism controls fore and aft movement.

On top of the engine casing an eye bolt is fitted at the suspension point of the centre of gravity, fulfilling two purposes. The first and obvious one is to provide a ready means of lifting the engine when out of the frame. The second is to furnish a simple and efficient means of engine removal and replacement. A simple hoist consisting basically of two roller chains and a square thread screw jack is fitted by a bracket to the right-hand side chassis frame member adjacent to the suspension point. It is intended to be operated by the vehicle wheel nut brace. A steady bracket for attachment to the body side is provided. Reference to the illustrations clearly



General view of engine installation.



Installation of hoist for removal and replacement of the engine.

shows the method of attachment and operation. A low-wheeled platform bogie aids the process of engine removal which can be effected by two fitters in 13 minutes. Refitting has been effected in 24 minutes.

Radiator and fan unit

Disposition of the radiator unit, its fan and drive, also the necessary connections can become quite difficult on a chassis of unorthodox design. In this case a convenient and effective location has been found immediately ahead of the front axle position in order to prevent mud being thrown on to the core by the front wheels.

To keep the unit below the level of the top of the frame, a separate header tank is employed. It is made in the form of a shallow rectangular box, and is fitted on top of the right-hand frame side member immediately forward of the front engine mounting bracket. The filler pipe projects to the off-side and is closed by a spring loaded hinged cap.

One connection is taken from below the right-hand side of the header tank to the front cylinder head, and another from beneath the left side of the tank to the right-hand side of the radiator header. The return pipe leads from the left-hand side of the lower radiator tank direct to the underside of the cylinder block. It has necessarily to be upswept to clear the front axle movement.

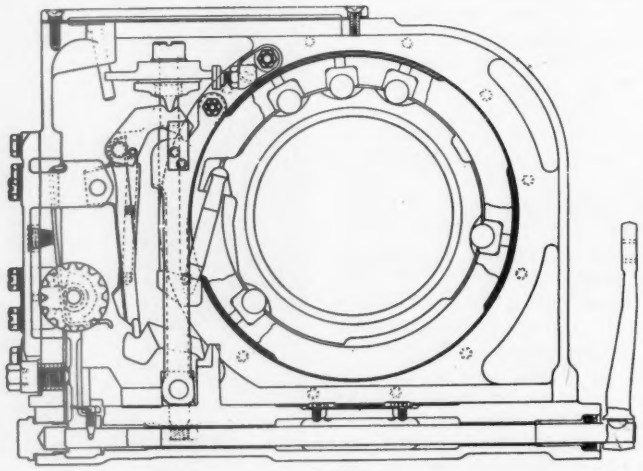
The radiator itself consists of a block built up of 74 Still-type tubes, having bolted to it cast aluminium alloy top and bottom tanks. A specially designed shroud to house the eight-bladed fan is fitted to the back of the radiator. The inner side incorporates vanes to promote correct air flow through the radiator and to prevent air cushioning due to swirl. This casting is machined around the fan path to ensure a close fit, and so give efficient operation. It carries the mounting for the fan which is driven by a shaft from the front of the engine.

The radiator is carried by brackets on either side at the top of the shroud casing, which rests on supporting brackets bolted inside the frame side members. The single attaching bolt on each side is spring-loaded in order to provide a degree of resilience and to avoid the possibility of wracking strains. Any tendency to longitudinal movement is restrained by a rubber bushed tie rod between the bottom of the radiator and the top of the frame cross-member behind the radiator. The total capacity of the cooling system is 12 gallons.

Fluid flywheel

The fluid flywheel is of the latest design in which the annular guide formerly used has been omitted and the circulating vortex is of oval section. A new feature is the fitting of an oil deflector plate to the central part of the driven flywheel member. These changes provide maximum freedom at idling speeds and minimum slip at working speeds, with consequent lower oil operating temperatures. The overall effect is a reduction in fuel consumption.

The flywheel consists of two main units, the casing which incorporates the driving member, facing forward, and the driven member which faces rearwards. Both the members are divided by radial webs into pockets, the driving member having 36 and the driven 39. The casing is in two main parts. The forward one is bolted to the driving flange of the crankshaft. On its outer forward edge it accommodates the starter ring gear. The rear member is secured to the forward one by bolts around the



Gearbox cross-section.

periphery. It houses a large diameter ball bearing in which runs the short, stout shaft that carries the driven flywheel member. The front end of the shaft is located in the tail of the crankshaft by a roller steady bearing. A lip type seal and a felt washer are fitted at the output end of the shaft in a housing which also holds the ball bearing in position. The rear end of the shaft is splined to take the flange of the forward Hardy Spicer coupling of the propeller shaft which carries the power to the gearbox.

Transmission layout

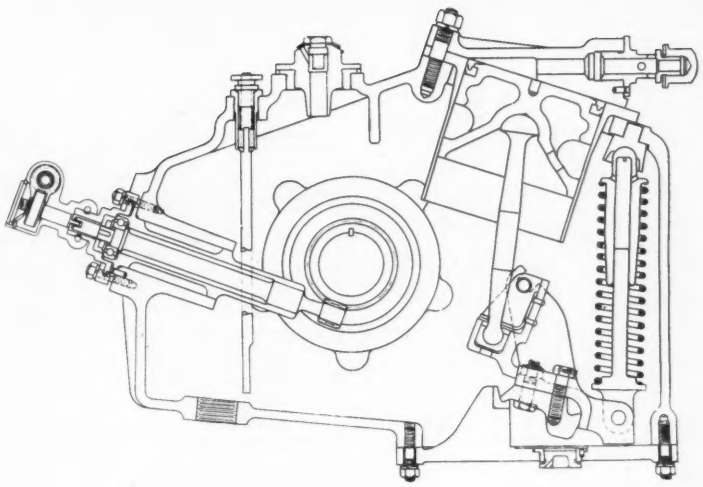
It will be seen from the chassis layout that the transmission is so laid out that engine, gearbox and rear axle drive points lie close to a straight line parallel to the right-hand frame side member. The two propeller shafts have Hardy Spicer needle roller bearing universal joints at each end. Very short shafts are fitted to vehicles for the home market, but they are necessarily longer on the overseas models.

Gearbox

Four forward speeds and a reverse are provided by the self-change epicyclic gearbox. This is mounted as a separate unit, its front end supported on a single mounting by the frame centre cross-member, and the rear end by two brackets from a special tubular cross-member. A feature of the gearbox is that it is operated by air pressure, the cylinder being built into the gearbox at the rear end.

In construction the gearbox follows accepted self-change gearbox practice, and in the main, the casing is a one-piece casting. The running gear is carried in substantial ball races mounted in the end covers of the casing. The separate base plate forms a mounting for the brake band assembly, the bus bar and part of the selector mechanism. The pre-selector camshaft mechanism is mounted on a plate which acts as a cover to the left-hand front side of the casing. There is a large cover at the top for inspection and adjustment. The internal mechanism consists of four epicyclic trains, each with its brake drum, and a multi-plate clutch that provides a direct drive top gear by locking the whole of the running gear.

The inner member of the top gear clutch is constrained about a flange formed integrally with the mainshaft of the box. Top gear thrust is also taken at the front of the box. Each brake drum, which is grooved externally to increase its holding capacity, has an external contracting brake band, lined with a friction material.



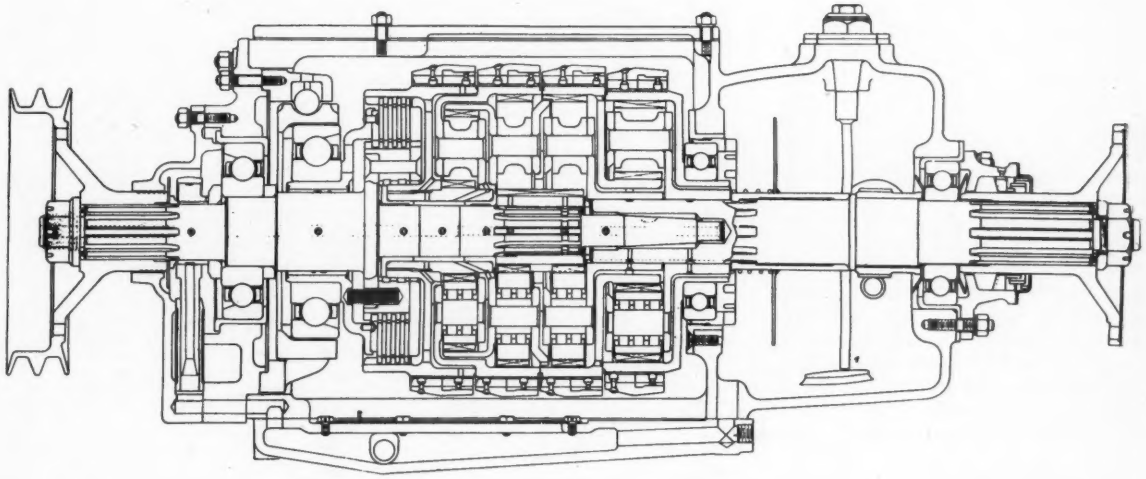
Air-operated gear control.

Air power is transmitted to the bus bar by a mushroom ended piston rod operating directly in the air piston. The layout can clearly be seen in the end section illustration, which also shows the skew gear drive for the speedometer take-off. The gearbox is lubricated by an oscillating plunger type pump at the front end, operated by an eccentric. Oil is forced through the drilling in the centre of the shaft, thence through axial drillings to the appropriate points.

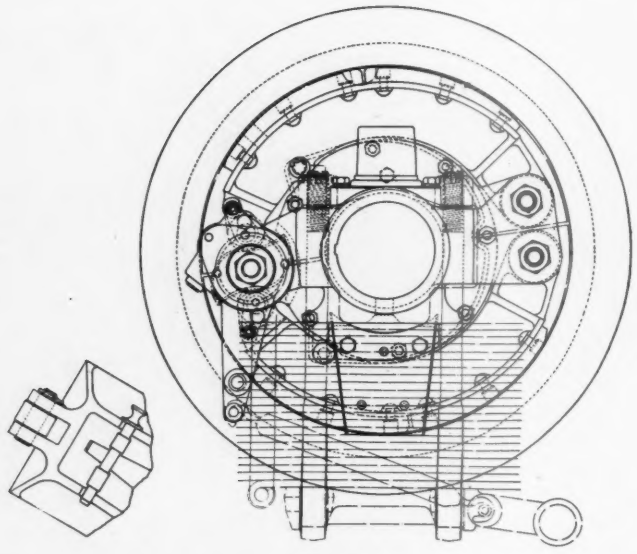
The gear ratios are:—

Top speed	direct
Third speed	1.64 to 1
Second speed	2.53 to 1
First speed	4.50 to 1
Reverse	6.9 to 1

The gear change control is mounted on the steering column, just below the wheel on the inner side according



General arrangement of pre-selective gearbox.



Rear brake assembly.

to whichever hand drive is employed. A form of gate is used in order to facilitate gear changing and to make it more definite.

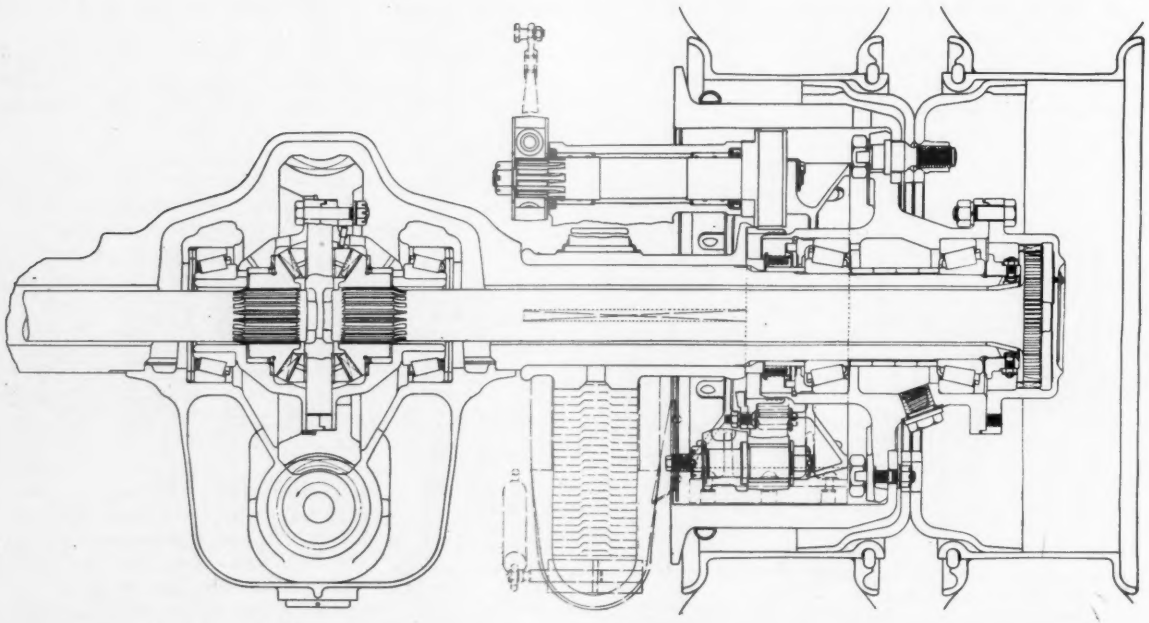
Rear axle

A fully floating underslung worm drive axle, offset to the right-hand side, provides the final drive. The one-piece axle casing is drop forged from alloy steel while the worm casing is in cast iron. A choice of four reduction ratios is offered: 4-1/7, 4-4/7, 5-1/6, or 5·8 to 1. Each is at 8in (203 mm) centres. An important new feature has been introduced so that the taper roller bearings, carrying the worm wheel and the four star differential, can be increased in size. Screwed adjustable housings have been superseded by shim adjustment in which it is possible to fit shims without having to remove the differential unit from its casing.

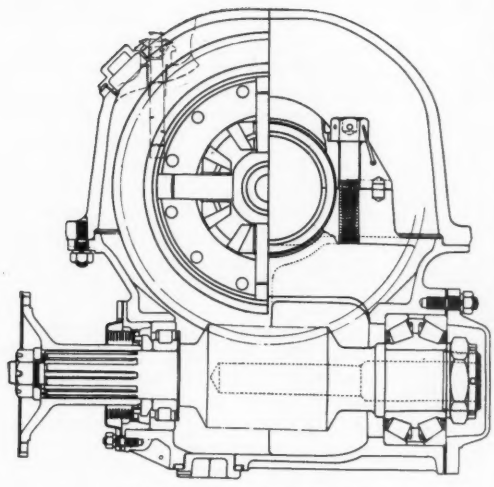
In order to ensure a positive drive and to relieve the securing bolts from shear load, an ingenious method of fitting the worm wheel to the right-hand differential casing has been adopted. The casing has a series of 156 teeth formed on about half the width of its outer diameter, thus forming a positive and accurate register on which the worm wheel is pressed. The worm shaft is made from alloy gear steel, hardened and ground. The worm wheel teeth are machined to a special profile to allow for the deflection which occurs under load. The differential bevel wheels and pinions are bushed and are provided with renewable thrust washers. To prevent oil leakage, a metal faced bellows-type packless gland oil seal is fitted to the worm shaft and to each hub.

Front axle

Of orthodox form, the front axle consists of a forged nickel steel "I" beam with integrally formed spring pads and well proportioned ends to carry the stub axles. The beam is inclined rearwards about 13 deg and is so arranged as to provide a high degree of centrepoint steering, which, with the unusual castor angle gives considerable self centering effect. The stub axles are mounted on taper centered, high tensile alloy steel swivel pins with phosphor bronze bushes at top and bottom. A synthetic rubber seal is fitted above the lower bush to exclude water and mud. Lubrication is effected from a grease nipple which, on the steering side, is on the steering arm, suitable connecting holes being drilled in both arm and stub axle eye. Thrust is



General arrangement of rear axle.



Section through worm drive.

taken at the lower end by substantial hardened and ground spherical thrust buttons, secured in place by a bolted-on cover.

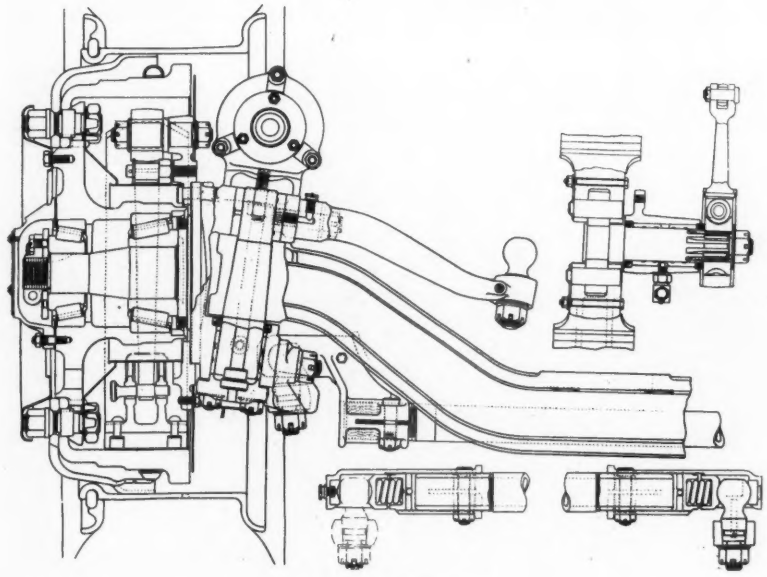
The steering arms are bolted on, and adjustable steering stops are provided, suitable abutments being formed on the axle beam. The hubs are carried on opposed Timken taper roller bearings, lateral adjustment being provided by a locking ring nut. A seal is mounted at the inner side of the stub axle and bears on the inner bearing cover plate. Over each swivel pin the brake operating cylinder is mounted horizontally, the piston rod being directly coupled to the brake cam actuating lever. A substantial tubular steering tie rod is fitted with screwed and adjustable stamped ends carrying renewable ball seats, the inner ones being spring loaded.

Wheels and tyres

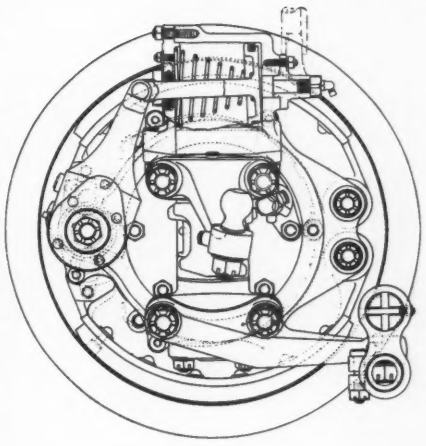
Dunlop standard ventilated steel wheels are used throughout, but size and tyre equipment varies according to the type of chassis. On the 7ft 6in wide home model, both front and rear rims are 20in diameter and 6in wide. 9·00-20 single tyres are fitted to the front and twins of the same size to the rear wheels. For 8ft wide vehicles 10·00-20 tyres, also on 6in rims can be fitted. On the 8ft wide model for overseas it is possible to fit a larger section tyre, and 11·00-20 tyres are fitted to 20in diameter, 7·33in wide rims.

Spare Wheel Carrier

In either case the wheels are secured in position by ten $\frac{7}{8}$in (22 mm) diameter studs per hub. A spare single wheel and tyre is normally supplied with each chassis on a carrier mounted on twin rails. A single screw holds the carrier in position. If the chassis is to carry a coach body, the carrier is fitted beneath the frame on the near-side, and if for bus operation, the carrier is fitted underneath the back of the chassis.



Arrangement of front axle.



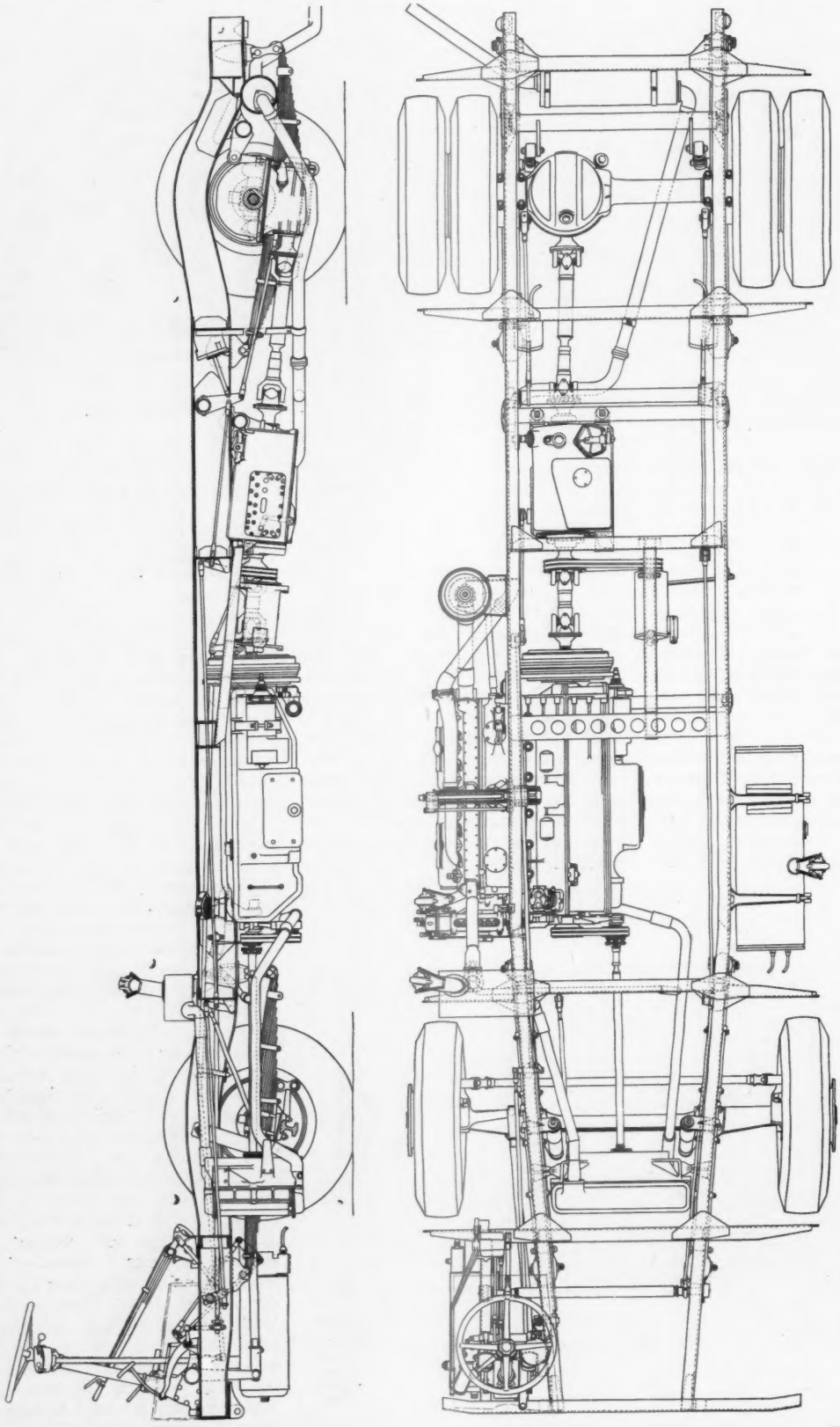
Front brake assembly.

Frame

A novel but straightforward design of frame has been made possible by the underfloor layout. Better balance has been achieved by extending the frame forward of the front wheels, the driving position being at the extreme front on a platform to right or left side as required. The general layout illustration shows the symmetrical balance of the design as a whole, the centre of gravity appearing to lie about the rear engine mounting. Parallel to each other for the major length of the chassis, the frame side members taper uniformly inwards from a point immediately in front of the engine to front cross-member. The frame has been kept as flat as possible, there being but a small amount of upsweep over both front and rear axles. The only other projection above frame top level, other than the steering column and driver's controls, is the small header tank for the engine cooling water.

Pressed from $\frac{1}{4}$in (6·35 mm) thick alloy steel, the channel section side members have a maximum section of 8in (203 mm) in depth with $2\frac{1}{2}$in (63.5 mm) flanges. A close fitting channel section flitch reinforcement is fitted within the frame side members extending from a point in front of the engine to behind the gearbox rear mounting cross tube. It is, of course, tapered at the end to avoid a sudden change of section with localized stresses. Seven channel section cross-members unite and brace the side members. They are extensively relieved by large diameter circular holes in the web of the channels. Of the outriggers on either side, the four adjacent to the spring anchorages are intended to support a body of integral construction. As an

THE A.E.C. "REGAL" MARK IV (Continued)



GENERAL ARRANGEMENT OF A.E.C. "REGAL" MARK IV CHASSIS.

alternative the conventional type of body employing cross-bearers can be accommodated. The frame is assembled by hot riveting, the only welding being that to secure the attachment plates to the outrigger pressings. Jacking pads are provided at the back of the chassis.

In the case of home buses there is a straight frame extension, while home coaches have a dropped extension.

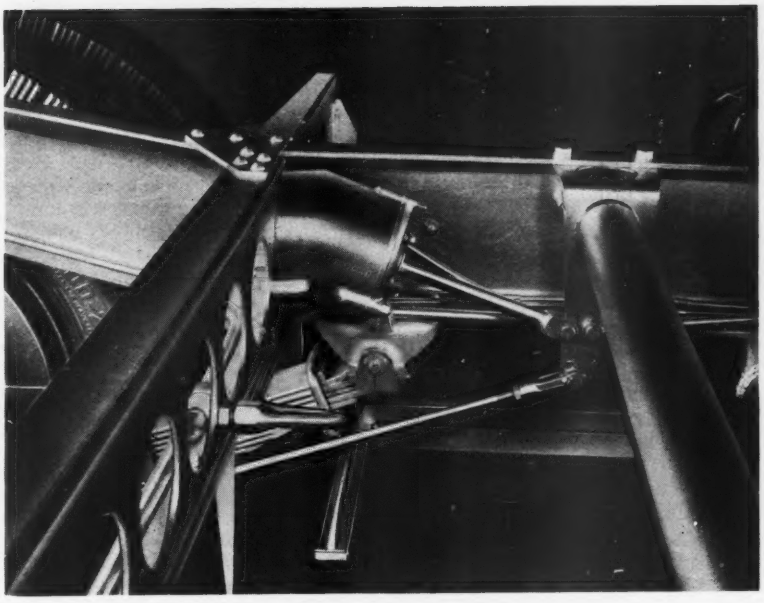
Suspension

Semi-elliptic, high deflection leaf springs, 5ft 2in (1,575 mm) in length, are used throughout. The reversed camber front springs have fifteen leaves and a full load deflection of 5¼in (133 mm). Rear springs have normal camber, have 22 leaves and a full load deflection of 6in (152 mm). In each case the anchorage is at the front of the spring and the shackle behind. Newton direct acting, tubular hydraulic dampers are fitted to the front axle and an A.E.C. patent stabilizer unit is fitted to the rear axle.

Brakes

Compressed air braking is employed, there being a separate operating cylinder for each wheel. Those for the front wheels are mounted individually over each stub axle and directly operate the levers on the brake camshafts. Those for the rear wheels are mounted on the frame cross-member immediately in front of the rear axle. These operate direct on the brake cross shaft levers, thence, through rods, the rear brake camshaft levers.

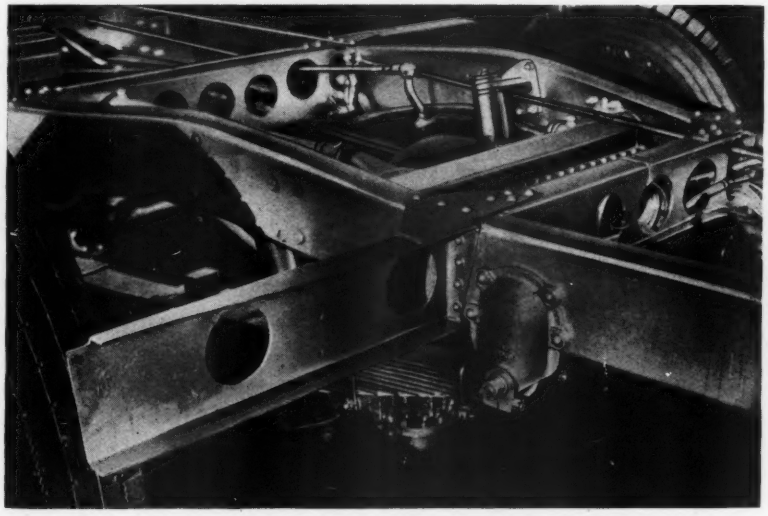
All four brake drums are of alloy cast iron and have an internal diameter of 15½in (394 mm). They are about 1in thick to aid heat dissipation and have deep, wide section flanges, to prevent distortion. The front shoes are 3⅝in (92 mm) wide and the rear shoes 6½in (165 mm) in width, the total effective area of the linings being 562 sq in (3,626 sq cm).



Rear brake cylinder, cross-shaft and linkage.

The brake shoes are fitted with rollers instead of the more usual flat cam plates, and long "S" shaped cams are employed. Worm and worm wheel adjusters permit the ¾in (19 mm) thick linings to be worm away fully without the necessity for the use of packing pieces. The linings which are secured to the shoes by riveting, are tapered in thickness when new so that as they wear they equalize in thickness. The shoes are held off by horse-shoe shape plate springs. The hand brake is mechanically operated by a rod to the rear brake cross shaft. The lever has a triple pawl ratchet.

Power for brake operation and for gear operation is provided by compressed air, from a 9.87 cu ft compressor built into the engine. The air reservoir, which may be of Westinghouse or Clayton manufacture, is mounted beneath the driving position according to the hand of drive. It incorporates the brake and gearbox air pressure operating valves so that the whole of the pressure valve gear is in a readily accessible position. Pipework is also simplified. Air mains are of ⅝in outside diameter, seamless pipe, having screwed connections with olive unions.



Rear spring attachment and body out-rigger.

Steering

A.E.C. worm and nut type steering is employed. A rocker shaft carries a bronze nut that swivels on two trunnions and is operated by a worm integrally formed on the bottom of the steering column. The column is located at the top by a ball bearing which also takes the vertical thrust. In the act of steering, the bronze nut passes through an arc which in turn causes a slight rocking action of the steering shaft. To accommodate this movement and to obviate any tendency to bind, the ball race is carried in its mounting, positioned by two rubber rings, which also absorb shock. The steering wheel is 21in diameter

THE A.E.C. "REGAL" MARK IV (Continued)

and is provided with a jack screw to aid removal from its taper. The drop arm depends vertically on the inner side of the column, thus permitting the side rod to take up a horizontal position parallel with the chassis centre line.

Fuel tank

Outrigger brackets, bolted to the side member, carry the fuel tank outside the left-hand side of the frame. Home models have a tank of 35 gallons (160 litres) capacity, while overseas ones have a tank holding 45 gallons (205 litres). The fill pipe projects to the left side and has a cast aluminium spring-loaded filler cap identical with those used on the sump and water header tank. The caps have an integral finger grip. They not only facilitate rapid servicing but prevent loss and afford the valuable feature that they need only be open for the minimum time, so essential in dusty operating conditions. A magnetic type fuel gauge is fitted.

Electrical equipment

Lighting and starting is by a 24-volt insulated return installation which may be of Simms or C.A.V. manufacture. Compensated voltage control is fitted, and the 7in (178 mm) diameter dynamo, which has a maximum output of 1,320 watts, is mounted on a swinging cradle and has a quadrant adjuster. It is driven by twin vee belts from a pulley mounted on the gearbox input shaft. 180 ampere-hour batteries of lead/acid type are carried in the body. The control box and battery isolating switch are mounted on the frame. The lighting switches and the starter button are mounted in the instrument panel, which is bolted to the front of the steering column close up to the steering wheel.

The instrument panel also carries the speedometer head, a two-pin plug for service light or portable lead, and indicator lights for dynamo, oil pressure and direction indicators. The panel is fitted with adjustable illumination. The horn push, head lamp dipper switch and direction indicator control switch are mounted on an outrigger bracket on the opposite side to the gear change lever, according to whether the chassis is right- or left-hand drive. The gear change is always to the inner side, therefore these controls are always to the outer side of the steering column.

Chassis lubrication

Lubrication nipples, connected by pipes to the appropriate points, are grouped in convenient points on the chassis frame. An engraved indicator label is mounted under each nipple. If desired, the R.P. automatic lubrication system can be installed as an extra.

CORROSION PREVENTIVE PACKAGING

IN a booklet recently issued by Shell Chemicals, Ltd., a new method of corrosion preventive packaging is discussed, with particular reference to the use of chemicals which are effective in the vapour phase. The action of a large number of organic compounds has been studied by Shell, and from this work evolved the vapour phase inhibitor known as Shell V.P.I. 260, which is claimed to prevent corrosion even when moisture and oxygen are present and which does not depend upon the inclusion of such corrosive influences to be effective. V.P.I. 260 is a stable crystalline organic compound which vaporizes slowly, giving a vapour which is odourless, non-toxic and non-injurious to the skin. It is soluble in water and alcohols.

It protects by its vapour, which completely surrounds any article in an enclosed space. Direct contact with the solid V.P.I. is not required. It is suggested that the vapour forms a very thin film over the metal surface, rendering it "passive," therefore protecting it against corrosion. Although moisture and oxygen are necessary for corrosion to take place, Shell V.P.I. 260 does not react with or remove either of them, but operates by inhibiting their corrosive action. Even when a film of moisture is present, as in a very humid atmosphere, V.P.I. still protects by dissolving in the condensed water and thereby making it non-corrosive.

It may be applied either as a powder, as a solution, as a coating on wrapping material or it may be embodied in a "cocoon" webbing solution. Protection by the use of a treated wrapping material will generally prove the simplest and most economical method of using V.P.I. In the booklet are illustrated many examples of treated and untreated test specimens. Most of these are of steel, although examples of glass, aluminium, copper and magnesium alloy are shown. V.P.I. coated wrapping paper is now being produced and supplied under licence from the Shell Organization by Leonard Stace, Ltd., Cheltenham. (1887)

INDUSTRIAL DIAMONDS

USERS of diamond tools, hard abrasives and sintered carbides will welcome the *Industrial Diamond Trade Names Index* which has been compiled jointly by the Industrial Diamond Information Bureau and the Industrial Diamond Review. It is in two sections, an alphabetical trade name index and a classified subject index. The alphabetical index includes the name and address of the firm responsible for the manufacture of the product. About 1,200 trade names are listed. This index is published by N.A.G. Press Ltd., 226, Latymer Court, Hammersmith, London, W.6, at 3s 6d per copy. (1872)

TIPPED TOOLS

BY carrying out a research into customer's requirements, the leading manufacturers of tungsten carbide tipped tools have found that certain sizes and shapes are now redundant. The present range of standard tools constitutes over 260 shapes of tips. By restandardization the number has been reduced to 116 without affecting the range of application. Many of the new tools have been designed with primary and secondary rakes for greater cutting efficiency and to allow tools to be employed on a variety of work with little modification to the rakes. It is intended that the new tools shall be available on and after 1st April, 1950.

For the benefit of users of Wimet tools, A. C. Wickman, Ltd., Tile Hill, Coventry, have produced a booklet that shows the equivalent new series tools against the old series which they will replace. This Company has also produced a new Wimet Standard Tools Manual. This manual contains a complete price list for the new series of standard tools. In addition it contains a wealth of information concerning the application of Wimet tools and is a publication of great value to all engineers concerned with machining problems. (1882)

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

The following meetings will be held during April:—

BIRMINGHAM CENTRE

Tuesday, 25th April, 6.45 p.m. General Meeting in the James Watt Memorial Institute, Great Charles Street. Paper: "Automobile Electrical Equipment for Passenger Cars" by M. W. Kendall, M.I.E.E.

NORTH-WESTERN CENTRE

Wednesday, 26th April, 7.15 p.m. General Meeting in the Grosvenor Hotel, Chester. Paper: "Some Factors Governing the Performance of Crankcase Lubricating Oils," by A. Towle, M.Sc. (Eng.), M.I.Mech.E.

WESTERN CENTRE

Thursday, 27th April, 6.45 p.m. General Meeting at The Royal Hotel, Bristol. Paper: "An Application of Petrol Injection to Automobile Engines," by R. Barrington, M.Sc., A.M.I.Mech.E. and E. W. Downing.

TYRE TESTING

An Important Development by The Avon India Rubber Co., Ltd.

ONE of the biggest problems in tyre design is the time lag between the introduction of a change in design and the final assessment of its effect on tyre performance. In addition, the actual "measurement" of performance is a complex matter. It involves many factors, all subject to considerable variation. For example, it is known that at high speed, tyre losses represent a very high proportion of engine power. It is also known that there are various methods for reducing these losses, but each method also has some effect on the other properties of the tyre. Consequently some degree of compromise is necessary, and the best compromise can be made only when the effects of any design change can actually be measured.

In considering the methods to be adopted for measuring tyre performance, it was necessary to decide whether the measurements should be made on tyres on test vehicles under actual running conditions or on tyres tested on a "laboratory" machine. After all the relevant factors had been carefully considered, The Avon India Rubber Co., Ltd., Melksham, Wilts, decided to adopt the second method. The chief reasons for this choice were that conditions could be controlled, tests could be repeated and the testing time could be very considerably shortened. This decision entailed the development of a special tyre testing machine that would be suitable for measuring tyre performance in scientific terms.

Conception of the test machine

A comprehensive review of all literature hitherto published on tyre stability showed that the measurement of tyre performance was based on:—

(a) Its power consumption.
(b) Its cornering power.
(c) Its self-aligning torque.

It was therefore decided that the new machine should be designed to measure these qualities under a wide variety of tyre conditions and should also be suitable for simulating operational duties on both front and rear wheels on road vehicles. To meet this second condition, the design had to allow for variations in king pin inclination, wheel offset and camber angle. Provision was made for the tyre conditions to vary in:—

(1) Speed.
(2) Inflation pressure.
(3) Tyre load.
(4) Slip angle.
(5) Driving and braking torques.

As the machine was to be used regularly for testing purposes, the design was further complicated by additional provisions that were considered necessary. These were to include:—

(1) Capacity to test a complete tyre range up to a maximum overall tyre diameter of 33in. A tyre in position for test is shown in Fig. 2.
(2) Easy tyre changing facilities.
(3) Simple and rapid methods of calibration for all instruments.
(4) Accessibility of all parts to simplify maintenance.
(5) Safety devices to protect the machine and tyre in the event of unforeseen failure occurring during test.
(6) Concentration of all controls and instruments in a special control room to allow one operator to carry out a complete test without difficulty.
(7) Room temperature control to eliminate variations arising from changes in temperature.

Constructional details of the machine

At the conclusion of the 1939/45 war, contracts were placed with Heenan and Froude, Ltd., Worcester, for the design and fabrication of a tyre testing rig from the first principles laid down by the Avon India Rubber Co., Ltd. It was known that a somewhat similar but much less elaborate machine existed in America, and a description of this machine had been published in the S.A.E. Journal in 1939. Nevertheless, the general and the detail design of the new machine had to be developed from fundamentals. Many design problems had to be solved, and many novel features were introduced, which have since proved valuable.

An illustration of the complete plant is given in Fig. 1. Essentially, the plant comprises:—

(1) A 50 g.p. variable-speed Heenan D.C. electric dynamometer, with speed-reducing gearbox, and capable of absorbing or delivering power.
(2) A steel drum coupled to (1). This has a diameter of 67.23in and thus makes 300 revolutions per mile.
(3) An "axle" with a universal joint at each end. One end is arranged to take the wheel and tyre on test, and the other is coupled to:—
(4) A second 50 h.p. Heenan D.C.

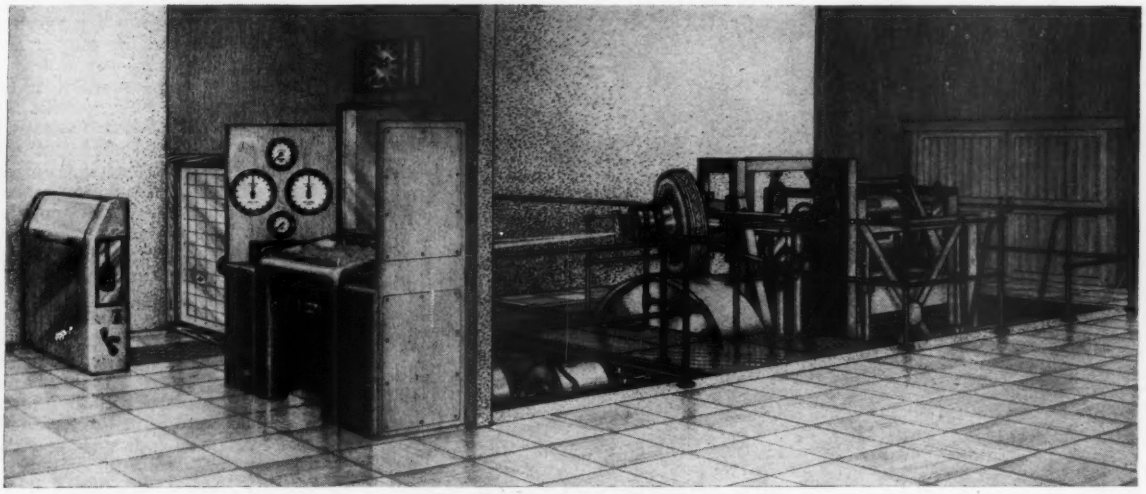


Fig. 1. The Avon India Rubber Co., Ltd. tyre research laboratory with the control room at the left and the stability test rig at the right.

TYRE TESTING (*Continued*)

electric dynamometer that is also capable of absorbing or delivering power. It is suspended by steel springs so as to be free to float axially, resisted by (5):—

(5) Hydraulic force-measuring gear, indicating axial thrust.

(6) Hydraulic force-measuring gear, by which a measured vertical force is applied to the axle and thence to the tyre.

(7) Arrangements for controlling and measuring the forces in the various mechanisms which affect the steering angle, dynamometer torques, etc.

(8) A main control panel in a separate room.

The speed-decreasing gearbox of the first dynamometer is mounted and yoked to the dynamometer carcass so that the torque-measuring mechanism truly indicates the net torque on entering the gearbox output shaft. This does not precisely correspond with the torque reaching the tyre, owing to drum bearing, and windage losses. However, a very close approach to these losses can be obtained by a process of calibration while the plant is running light, that is, without a tyre in place.

The second dynamometer, complete with its supporting cradle and thrust-measurement gear, is mounted upon a structure that embodies facilities for considerable vertical adjustment. This allows the machine to be used for a range of tyre diameters. The speeds of rotation of each dynamometer are such as to allow tyres to be tested at any desired linear speed between 30 and 120 m.p.h.

In order that the axle which carries the test wheel may be restrained from sideways motion, but may be free to move axially and vertically, it is provided with a special assembly of roller guides. These are, of course, so arranged that they do not interfere with the transmission of thrust to the thrust measuring equipment.

The load-imposing gear for controlling the vertical load on the tyre is, in principle, an inverted replica of the hydraulic equipment used by Heenan and Froude Ltd. to measure the torque forces in very large dynamometers. By movement of a small weight to an appropriate position along a steelyard in the control room, any desired force can be applied by a hydraulic ram connected to the axle, see Fig. 3, and maintained constant at this value. When a small tyre is under test, this force may even be negative, in order partly to counteract axle weight. Should the tyre fail under test, the consequent downward movement of the axle causes contact to be made with an adjustable trip mechanism. As soon as this contact is made, the hydraulic forces are reversed and the ram lifts the wheel and tyre clear of the drum. The object of this is to preserve as far as possible the evidence of the cause of tyre failure.

Hydraulic capsules are used for measuring the forces due to axial thrust, camber, dynamometer torque and self-aligning torque. They are connected to large diameter pressure gauges, and where necessary are arranged to accommodate movement. For example, the self-aligning torque gauge is mounted as part of the swinging arm by which the wheel on test is steered. Variations in king pin inclination are made by causing the entire axle to slope, that is, by raising or lowering the second dynamometer assembly complete. A separate adjustment controls camber angle. All tests are carried out at zero caster.

The construction of the axle itself is of special interest. It has a non-rotating outer tube, provided at one end with a universal swivel on the centre-line of the king pin axis. Inside the outer tube is a co-axial rotating shaft through which the torque is transmitted. Each end of this shaft is fitted with a Bendix-Weiss universal joint having a centre of oscillation corresponding with that of the outer shaft swivel joint. Weiss joints were chosen partly because of their small dimensions in relation to duty, and partly because of their constant-velocity properties.

At the outer end of the axle there are bearings which support a rotating hub that is driven from the inner torque shaft. This hub terminates in a flange to which various rim sections may be bolted. By an arrangement of screw threads, the position of the flange may be adjusted axially, the main object being ability to alter "offset." The hub dimensions are necessarily limited to suit the smallest size of tyre that will be tested. As a result, considerable ingenuity was necessary to accommodate the various mechanisms within the hub.

At the outer extremity of the hub there is a small non-rotating mirror, from which a beam of light is reflected on to a large semi-transparent screen in the control room wall. This screen is shown immediately to the left of the control panel in Fig. 1. Variations in the horizontal and vertical angles of the hub, due respectively to changes in the steering angle and the camber, cause the beam to be displaced on the screen, where they can be measured with a high degree of accuracy. Such accuracy is necessary because, for example, small changes in steering angle cause relatively large changes in forces. The mechanisms for varying the steering angle and the camber are shown in Fig. 4.

A special valve and seal are mounted integrally with the hub so that, if necessary, compressed air can be fed to the tyre during the course of the test. This allows the pressure inside the tyre to be altered at will to reproduce any desired conditions. Generally, this arrangement is used to maintain a constant air pressure in the tyre throughout the duration of the test.

In addition to the axle load indicator and the large screen referred to earlier, the control room also houses a central control panel, carrying four pressure gauges which respectively indicate self-aligning torque forces, cornering force and dynamometer torque forces. There are also two speed indicators and the controls for the two dynamometers.

Naturally, many difficulties have been encountered, not only in the operation of the plant but also in the design of the tests and in the analysis

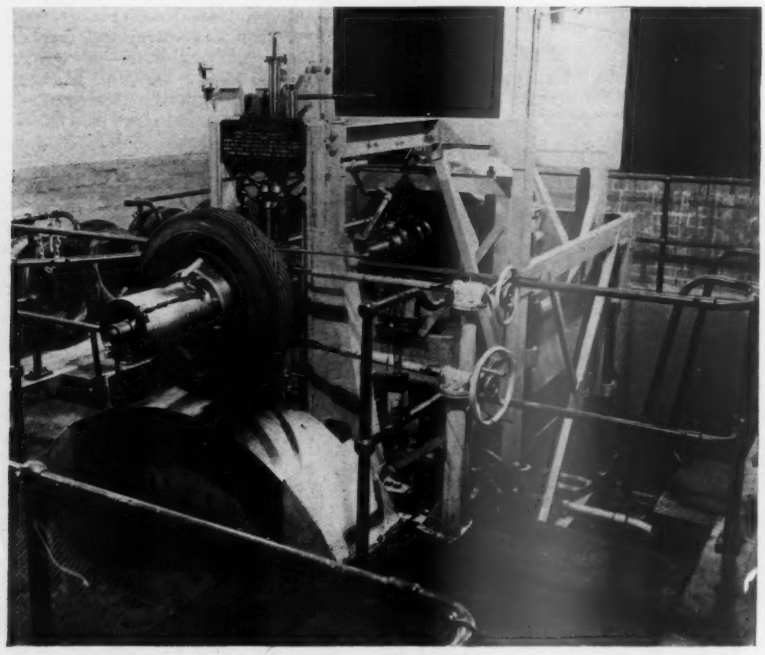


Fig. 2. A tyre set up ready for test.

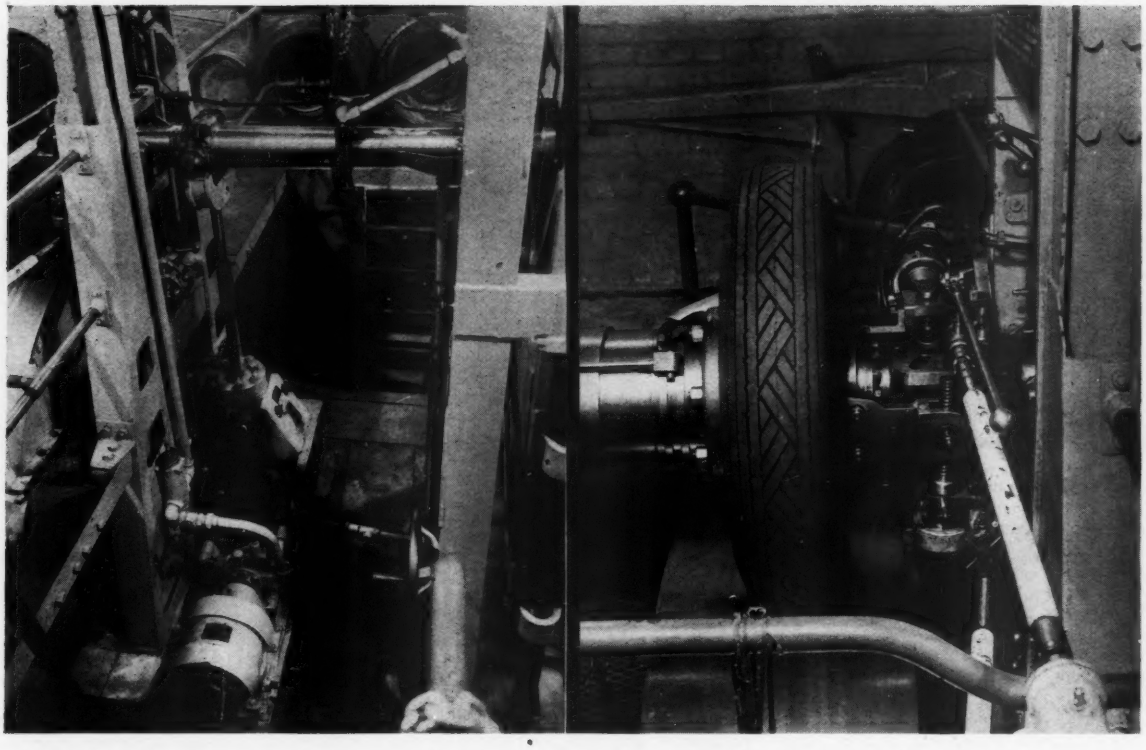


Fig. 3. Hydraulic mechanism for adjusting the axle inclination and for imposing the load on the tyre.

Fig. 4. Mechanism for varying the steering angle and the camber. Alteration can be made while the machine is running.

and interpretation of the results. This was only to be expected with a unique piece of equipment. However, the majority of the problems have been solved and the machine is now beginning to fulfil its intended role as an accurate measuring instrument.

With this machine the correlation of cause and effect can be approached in a scientific manner that was previously impossible. In a few days, or sometimes even hours, the effect on tyre performance of the smallest change in design can be assessed in exact terms. There is no doubt that this machine is an outstanding development of great importance to both the motor industry and users of road vehicles. The combination of controlled conditions for a wide range of tests and accurate measurement of the complex forces involved must play a great part in the constant endeavour of The Avon India Rubber Co. Ltd. to produce improved tyres to keep pace with the increasingly exacting demands of modern road vehicles.

LATHE DEVICES

THREE new devices have recently been developed by Alfred Herbert, Ltd., Coventry, for use with the Company's capstan and turret lathes. The first is a chip guard designed to give the operator complete safety from flying chips when the machine is running at the high speeds that are possible with Ardoloy and other cemented carbide tools. Although the guard eliminates the danger from flying chips, it allows a clear view of the work through an armour-plate glass window. As it is mounted on rollers running on a bar of H section rigidly bolted to the headstock of the machine, it is easily manipulated. A wire mesh guard at the rear prevents chips from flying in that direction.

Spacing Device

The second device is an accurate yet simple spacing bar for fitting to Herbert turret lathes. When mounted on the stop bar, which is normally provided with automatic trips and dead stops for the longitudinal motion of the saddle, the bar provides for accurate spacing of grooves or similar recesses. It eliminates the necessity of indexing the stop bar and resetting the trip stops when the number of grooves required is in excess of the number of stops available. This arrangement does away with the need for expensive special attachments, using spacing blocks. It is also much more accurate and reliable than the alternative method of marking the saddle traversing handwheel, since such a method depends upon the human element.

Adjustable Lathe Tool

To facilitate rapid and accurate setting of lathe tools to the required centre height, Alfred Herbert, Ltd., have developed a new-type adjustable tool. It is set by means of transverse serrations in each of the angular faces of the tool shank. The serrations are accurately produced so that the maximum rigidity is assured when the two-piece tool is clamped in the tool post. These serrations allow adjustment in increments of 0·005in. With this tool setting-up, particularly after regrinding, is much easier than when the conventional method of using shims is employed. (1870)

THE SKEFKO BALL BEARING CO., LTD., Luton, have published a souvenir booklet to celebrate the 40th anniversary of the founding of the Company in England. Brief historical details are given along with a large number of illustrations taken at various periods of the Company's existence. Interesting facts are given concerning the organization, together with many examples of the firm's products.

NEW PLANT AND TOOLS

Recent Developments in Production Equipment

A RANGE of double-ram horizontal surface broaching machines has recently been added to the range of equipment manufactured by The Lapointe Machine Tool Co. Ltd., Edgware, Middlesex. These machines incorporate several novel design features and are intended specifically for high-production requirements such as those met with in the automobile industry. Under actual production conditions and a cutting speed of 25ft per minute, one complete connecting rod and cap can be produced from the rough forging.

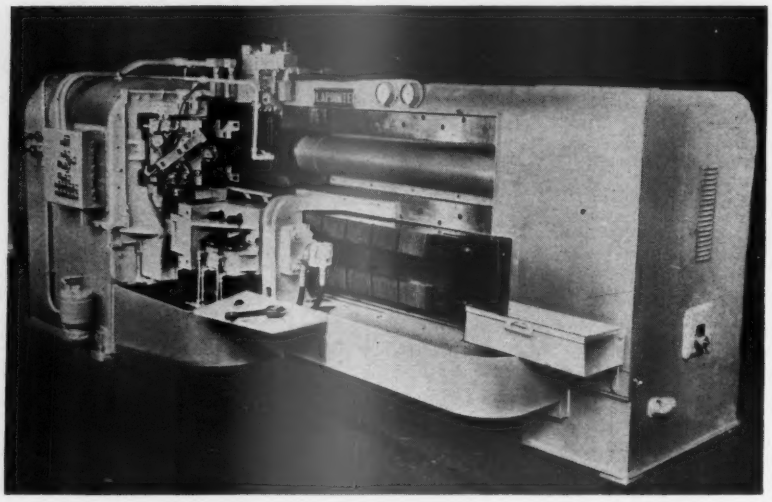


Fig. 1. Lapointe horizontal broaching machine.

These machines are available in either 15-ton or 25-ton capacity with 66in, 90in and 120in stroke. A 15-ton model is shown in Fig. 1. Because of its relatively low height the machine is particularly suitable for use where the headroom is limited. Wherever possible the mechanism is neatly enclosed. In addition, ingenious guards travel with the broaches on both the cutting and return strokes so that the tools are completely enclosed and the risks of accident to the operator or damage to the machine are made practically negligible. On the 15-ton model the speed is infinitely variable between 5 and 21ft per minute, and between 5 and 16ft per minute on the larger model. Where the machine is subject to very heavy operating conditions, an automatic cooling device may be fitted to the hydraulic system.

A novel design feature that is of particular value for the automobile industry is the provision of universal broach inserts mounted in adjustable sub-holders, see Fig. 2. This arrangement allows up to seven similar, but different size, components to be machined with one set of broaches. It applies to both the upper and the lower sets of broaches. As a consequence, to change over from one size of component to another, it is only necessary to adjust and not to replace the broaches. This saves considerable setting-up time. In addition, it reduces the tool investment, since one set of tools will suffice for work that would otherwise require several sets.

Another design refinement is the provision of a dual-control panel that is within easy reach from the normal operating position. Individual push-buttons, mounted on the face of the panel, allow any machine movement to be controlled independently of the other machine movements. This is a great convenience for setting-up. For operational convenience, the push-buttons are arranged in a logical sequence. This does away with need for constantly moving backwards to study the position of the buttons. On the front edge of the panel is a second set of push-buttons for controlling the automatic cycle. When one set of controls is in use, a full electrical interlock system renders the other set inoperative.

Special work-holding fixtures, see Fig. 3, are available to suit either full or semi-automatic operation. Naturally, the design varies to suit the type of work to be broached, but they are all built with fully-interlocked swinging-type tables of heavy section. Quick-action locking devices reduce loading times to a minimum, and the tables can

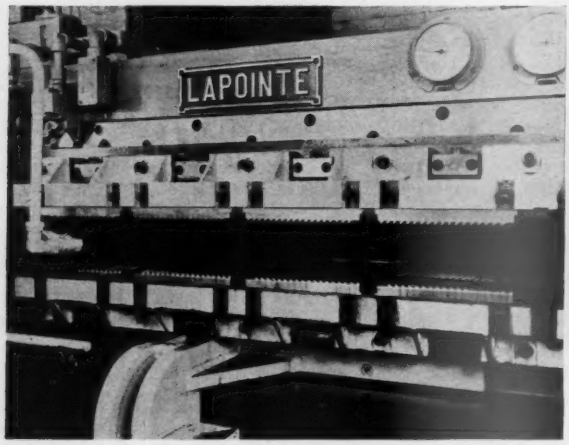


Fig. 2. Broach body and the method of adjustment on the Lapointe horizontal broaching machine.

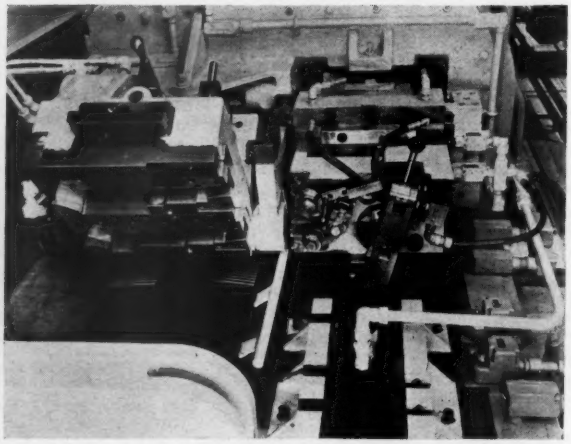


Fig. 3. Details of the lower work fixtures on a machine tooled for broaching connecting rods.

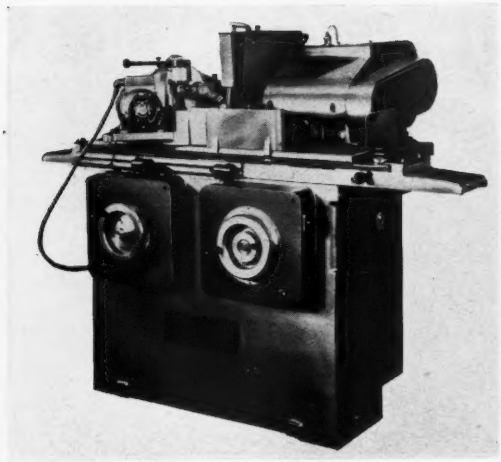


Fig. 4. Jones and Shipman valve grinding machine.

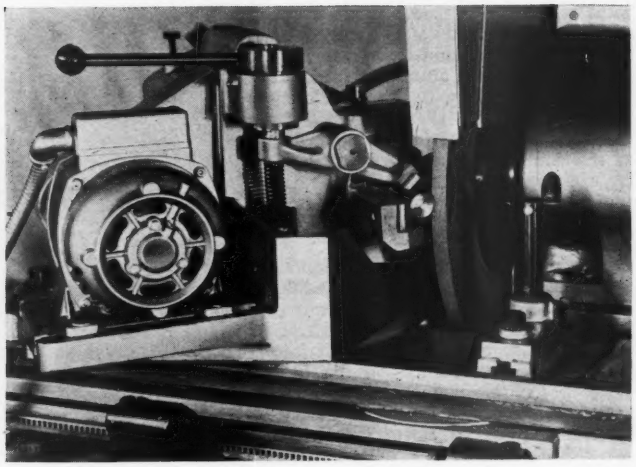


Fig. 5. Method of holding the valve to give eccentric grinding.

be swung clear to facilitate insertion and removal of the work.

Valve grinding machine

A new design of poppet valve grinding machine developed by A. A. Jones and Shipman Ltd., Narborough Road South, Leicester, is illustrated in Figs. 4 and 5. It is equipped with a special type of work-head that has a device arranged to ensure that the seats are ground true to the stem of the valve, and it has a wheelhead with an oscillating spindle. The workhead has been specially designed to rotate the valve in such a manner as to ensure that the finished seat is true with the stem. This is effected by gripping the extreme end of the valve in a collet for rotation, but locating the remainder of the valve stem up to the head in a vee. As the stem locating device is in the form of a vee the pressure of the grinding wheel always tends to retain the valve in the bottom of the vee. A top steady is also included. This also presses the valve into the vee and prevents any possibility of bounce or chatter.

The collet for gripping the valve stem is built into the workhead spindle, and the lever controlling the collet mechanism is coupled to the mechanism that operates the top steady. To load a valve into the workhead it is only necessary to operate this one lever and push the valve through the vee into the collet. Correct location is given by contact between the end of the valve stem and a stop inside the collet. The workhead spindle is made of Nitralloy steel and runs in phosphor-bronze bearings of ample size. To allow for grinding valves of different angles, the workhead body is mounted on a swivel base.

A wheel 18in dia. × ¾in wide is mounted on the wheelhead spindle. This spindle is also of Nitralloy steel. It runs in phosphor-bronze bearings which are constantly lubricated from a positive delivery pump. The pump draws oil from a reservoir inside the wheel casing. The wheel spindle is arranged for continuous side oscillation. This allows components to be fed directly into the wheel for plunge grinding. This feature ensures a high degree of finish on the valve seat, and serves to give longer wheel life by avoiding the possibility that grooving of the wheel will occur. Traverse and cross feed controls are arranged at the front of the machine in positions convenient for the operator, and adequate guards are provided to protect the operator from spray.

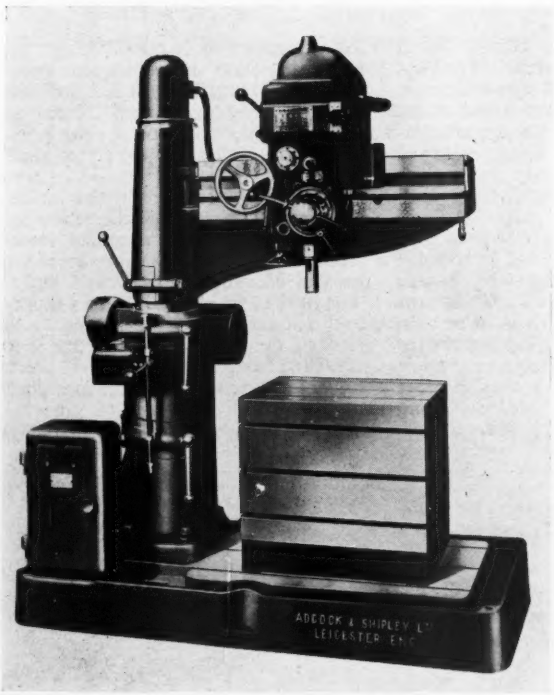


Fig. 6. Adcock and Shipley 3ft. radial drill.

Radial drilling machine

Adcock and Shipley Ltd., Ash Street, Leicester, have introduced a modified and improved version of their "SV" model radial drilling machine. The new model is illustrated in Fig. 6. It has a capacity of 1in. in steel, and is of the low base type which can be supplied with either a standard box table or a swivel table.

The arm, which is designed to withstand very heavy stresses, is mounted on a cast iron column which can move vertically in an outer cast iron sleeve. Elevation and lowering of the column are carried out through a rack and pinion movement operated either by electric motor or by hand. On the power-operated model there is a bar with adjustable trips to stop the motor at predetermined positions. The method adopted for adjusting the relative height of the arm and the work gives great rigidity. In fact, tests have shown that the deflection of the arm is as little as one-third of the accepted standards.

The saddle runs on vees and flats on the arm. It carries a four-speed, pole-changing motor stator unit built in at the rear and driving the spindle through vee ropes. In conjunction with a back gear, the four-speed motor gives eight spindle speeds, ranging from 96 to 1,455 r.p.m. Reversing of the spindle for tapping operations is controlled through a conveniently placed lever.

In addition to the normal hand feed, there is a slow hand feed operated by a handwheel. There are also three

rates of automatic feed. When automatic feed is desired it is engaged by movement of the hand feed lever. Disengagement of the automatic feed can be arranged at any desired predetermined depth. A dial graduated in fractions of an inch facilitates setting of this control. The spindle is balanced by a clock-type spring, the tension of which can be readily adjusted. It runs in ball bearings in a quill that has an internally cut feed rack.

Powerful locks are provided for all movements of the arm and the saddle. To ensure long life for the moving parts, the saddle is pump lubricated. The electrical equipment is built into a neat cabinet mounted on the base near the column. A coolant sump is incorporated in the base, and coolant is circulated by an electrically driven centrifugal pump.

Surface grinding machine

A new type Thompson Hydrail surface grinding machine has recently been installed at the works of British Timken Ltd. It is one of the widest surface grinding machines ever made and it has remarkable production capacity. For example, it is doing in two hours, work that previously took three days and is grinding to a tolerance of 0.0008in over an area 72in × 120in.

The machine, which is illustrated in Fig. 7, is approximately 35ft long, 16ft wide and 12ft high. It will accommodate work up to 75in wide × 36in high × 120in long. Both automatic push-button and manual controls are provided. For operating convenience, the controls are duplicated on both sides of the machine. This machine can be supplied with the wheelheads built horizontally or vertically, or both if two wheelheads are desired.

Each grinding wheel spindle is driven by a 30 h.p. 1,200 r.p.m. motor. The spindles are hardened and ground and are carried in pressure lubricated, steel back, silver lined bearings of a patented type. An unusual and very valuable feature of the design is that the electrical system is so interlocked that the spindle will not run until a specified lubrication pressure has been built up in the bearings. Normally 24in dia. × 6in wide grinding wheels are fitted. Power for supplying the lubrication is derived from a 1/20th h.p. motor for each spindle.

There is 8in adjustment for raising or lowering the wheelhead on the rail. The mechanism is graduated for manual adjustment in increments of 0.0005in. Rapid power traverse can be supplied if desired. Provision is made for both manual and hydraulic cross feed of the wheelhead. Manual cross feed can be effected in increments of 0.0005in. The hydraulic cross feed is variable in the range from ¼in to 3in per table reversal. In addition, there is continuous rapid traverse variable between 10in and 50in per minute. Rail raising and lowering is effected through an electrical power brake motor at the rate of 24in per minute.

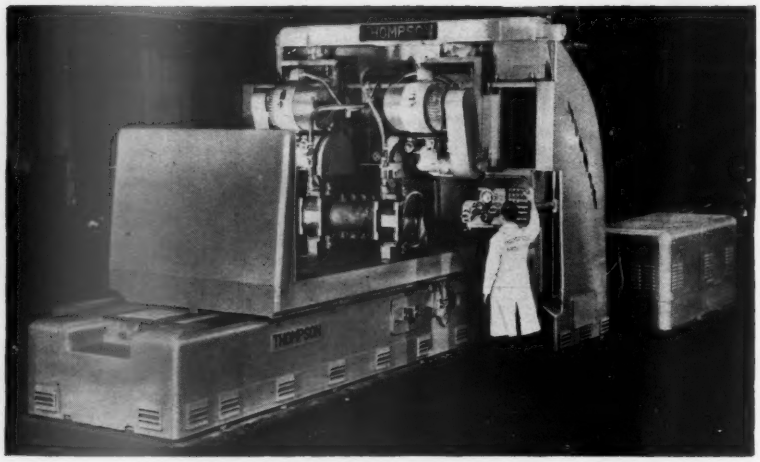


Fig. 7. Thompson Hydrail surface grinder.

The table is hydraulically operated. Power is derived from a 30 h.p. 1,200 r.p.m. motor and the table speeds are infinitely adjustable between 10ft and 30ft per minute. The bed and table ways consist of one 10in wide 90 deg vee bearing and a 10in wide flat bearing. These bearings have automatic pressure lubrication, and the system is so arranged that the hydraulic pump for operating the table will not start until the specified lubricant pressure is built up.

To give maximum rigidity even under extreme conditions, the length of the bed way bearings is twice the table stroke. In addition, they are completely sealed in all positions by Thompson patent Tapeway bed covers. All bearings on both columns, cross rail and vertical head are also completely sealed in all positions.

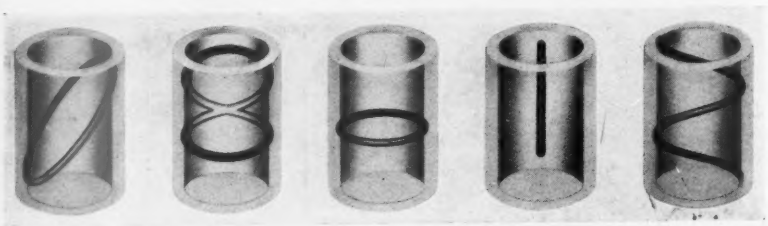
The machine is arranged for wet grinding and includes all the necessary equipment. There is an independent coolant supply to each head. Rockwell Machine Tool Co. Ltd., Rockwell House, 2nd Way, Exhibition Grounds, Wembley, Middlesex, are the agents for these machines.

STANDARD BRONZE BUSHES

The Glacier Metal Co., Ltd. Offers 191 Sizes

SOME time ago the Glacier Metal Co., Ltd., Alperton, Wembley, Middlesex, instituted a standardization scheme that has so far resulted in the marketing of a standard range of steel-backed, lead bronze-lined, wrapped bushes, a range of strip type white metal-lined half bearings, and a standard range of bronze bars for those who machine their own bearing products. The Company has introduced another standard range of bronze bushes. These are offered in 191 standard sizes from ⅛in bore to 2½in bore with a choice of 4 to 7 lengths for each bore size. The bushes are made of lead bronze for which the following advantages over the harder bearing bronzes are claimed: (a) ability to conform to minor shaft misalignment, (b) reduced friction and consequent lower running temperature, (c) reduced rate of shaft wear, (d) reduced liability to seizure under conditions of poor lubrication, (e) ability to use softer shaft materials.

Regarding tolerances on the bushes as supplied, the outside diameters are over-size to provide adequate interference fit when the housing bores are held to Newall B or B.S.164U limits. The bushes are supplied with a finishing allowance in the bores. As stocked, the bushes are plain, that is to say, they have no grooves or oil holes. Oil holes and grooves can be machined to order and five types of oil grooving can be provided. (1883)



Oil groove patterns for Glacier standard bushes.

TRACKED VEHICLES

An Analysis of the Factors Involved in Steering

By W. Steeds, O.B.E., B.Sc., M.I.Mech.E.

SEVERAL papers have been published on the subject of the steering of tracked vehicles and there is a good deal of material relating to it that has been circulated only within the limits of government departments. There has not however, been any article dealing with the subject comprehensively and in a logical sequence. From the point of view of steering, tracked vehicles may be divided into two groups: (1) those employing laterally flexible tracks, and (2) those with laterally rigid tracks. A laterally flexible track is one that can be laid down on level ground in a curve while a laterally rigid track is one that can be laid down only in a straight line.

Since tracks must obviously be flexible in a vertical plane in order that they may pass round the sprockets and idlers the laterally flexible track has two " degrees " of flexibility and this necessitates a connection between the links that will permit rotation about a horizontal and a vertical axis, assuming the links to be in a horizontal plane. Thus a double pivot or universal joint type of connection is necessary but a ball and socket joint, which allows an additional and unwanted degree of rotation, maybe used.

A track employing such joints is expensive to produce, which is a very serious drawback. It has been found in the past to wear more rapidly and to be more prone to coming off the sprocket than the laterally rigid track. When a laterally flexible track is used steering is effected by laying the track down on the ground in a curve. This may be done in two ways both of which have been used, one in an experimental and the other in a production vehicle. In the experimental vehicle the track idler, which was at the front of the vehicle, was carried on a bracket pivoted to the hull on a vertical axis as indicated in Fig. 1.

When the bracket is turned through an angle on its pivot each track link as it is laid on the ground lies at an angle θ to the preceding link and so the track is ultimately laid on the ground in the arc of a circle. If P is the pitch of the track links then the radius of the arc in which the track is laid is

$$R = \frac{P}{2 \tan \frac{\theta}{2}}$$

The idler bracket on the inside of the turn will have to be turned through a larger angle than that on the outside.

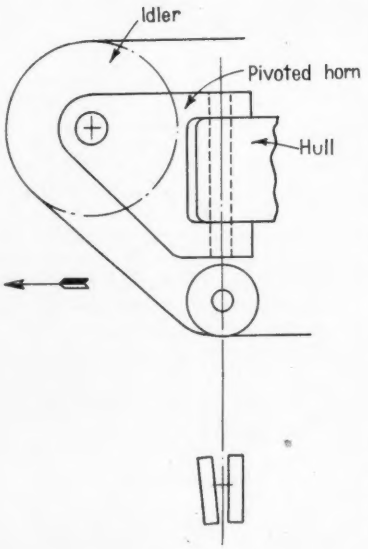


Fig. 1.

Assuming a pitch of six inches then a radius of ten feet corresponds to an angle of two degrees fifty-two minutes. Thus very small amounts of lateral flexibility will suffice.

The weight-carrying wheels of the vehicle must be arranged so that they can conform to the curvature of the track and this introduces an additional complication into the design problem. Also a pivoted idler will not give any steering in reverse and either the sprockets must be carried on pivoted brackets or other means must be used to steer in reverse, and this means more complication. In the production vehicle the track was laterally flexible only to the extent permitted by the clearances between the track pins and their holes in the track links, and the curved track method of steering was used only for large radius turns. The suspension of the weight-carrying wheels, which included the idler and sprocket, was such that the wheels could be turned through a small angle and moved bodily outwards or inwards so as to constrain the tracks to lie in a curve. The principle of this suspension is shown in Figs. 2 and 3.

When the curved track method of steering is used a differential will be incorporated in the transmission of the vehicle. The main advantage of steering by curving the tracks is that much less power is used in making the turn and this, presumably, means that less damage will be done to the road surface and that lower stresses and less wear should occur in the tracks and suspension system. The method would seem to be useful for light high speed vehicles but of very little use for heavy vehicles. The notes that follow, relate to vehicles having laterally rigid tracks.

The kinematics of steering

Whatever the actual steering systems there are, with laterally rigid tracks, certain kinematic relationships which always apply and these will now be developed.

Consider a track as shown in Fig. 4, in which the chain dotted rectangle indicates the hull that carries the track; suppose the hull to be turning about the *centre of turn* O with an angular speed Ω, and let the radius to the centre point of the hull be R. Then the mean speed V of the vehicle is given by,

$$V = R\Omega \quad .. \quad .. \quad .. \quad (1)$$

Supposing the shaded rectangle to represent the portion of the track lying on the ground then the angular speed of this track is also Ω because it must turn with the hull. If the track has any velocity relative to the ground then that velocity must, for the point Q_0 be in a direction perpendicular to OGQ_0. Suppose the track to be driving the vehicle forwards and to be itself slipping backwards over the ground at a speed, for the point Q_0, of v_0. Then the instantaneous centre of the track relative to the ground must lie on OG because if it did not do so the velocity of the point Q_0 relative to the ground would not be in the right direction.

Suppose the instantaneous centre to be

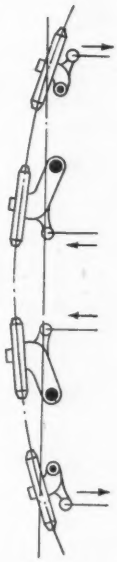


Fig. 2.

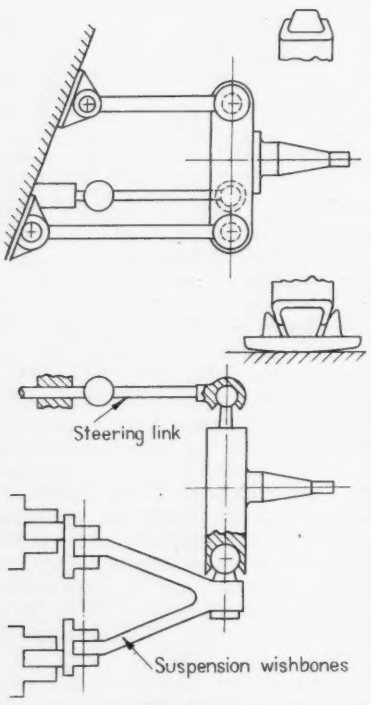


Fig. 3.

the point O_0 lying at the distance a_0 from Q_0. Then since the angular velocity of the track relative to the ground is Ω the position of the centre O_0 must be on the opposite side of the track from the centre of turn O, as shown, in order that the velocity of the point Q_0 may have its proper sense, that is, backwards, according to our assumption. The velocity v_0 may be called the *slip velocity* and the radius a_0 the *slip radius* and clearly we have the relationship,

$$v_0 = a_0\Omega \quad .. \quad (2)$$

The slip radius will be taken as positive when the instantaneous centre O_0 lies on the opposite side of the track to the centre of turn O. If the track slips forward relative to the ground then its slip velocity is negative and its slip radius is negative so that the instantaneous centre will lie between the track and the centre of turn.

The speed of the point Q_0 of the *hull* is clearly $(R + C)\Omega$ and the speed of the point Q_0 of the *track relative to the hull* is $(R + C)\Omega + v_0$; but this is equal to $r\omega_0$ where r is the sprocket radius and ω_0 is the speed of rotation of the sprocket about its axis. Hence we have the equation,

$$(R + C)\Omega + a_0\Omega = r\omega_0 \quad .. \quad .. \quad .. \quad (3)$$

For the inner track, assuming it also to slip backwards relative to the ground, the corresponding equation is

$$(R - C)\Omega + a_i\Omega = r\omega_i \quad .. \quad .. \quad .. \quad (4)$$

In actual practice the inner track usually slips forward relative to the ground so that the slip radius a_i is usually negative.

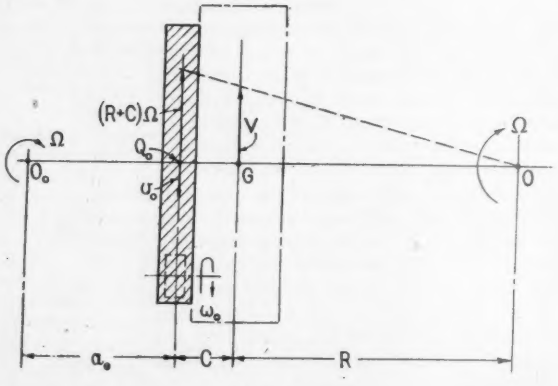


Fig. 4.

Dividing equation (3) by equation (4) gives,

$$\frac{R + C + a_0}{R - C + a_i} = \frac{\omega_0}{\omega_i} = x \quad .. \quad .. \quad .. \quad (5)$$

from which we get,

$$R = \frac{C(x + 1) + a_0 - a_i x}{x - 1} \quad .. \quad .. \quad .. \quad (6)$$

Other relationships that follow are,

$$R = \frac{V(2C + a_0 - a_i)}{r(\omega_0 - \omega_i)} \quad .. \quad .. \quad (7)$$

$$V = \frac{r[C(\omega_0 + \omega_i) + a_0\omega_i - a_i\omega_0]}{2C + a_0 - a_i} \quad .. \quad (8)$$

$$\Omega = \frac{r(\omega_0 - \omega_i)}{2C + a_0 - a_i} = \frac{r(\omega_0 + \omega_i)}{2R + a_0 + a_i} \quad .. \quad (9)$$

$$\omega_0 + \omega_i = \frac{\Omega}{r}(2R + a_0 + a_i) \quad .. \quad .. \quad (10)$$

$$\omega_0 - \omega_i = \frac{\Omega}{r}(2C + a_0 - a_i) \quad .. \quad .. \quad (11)$$

Special cases. Equations (1) to (11) are true for all values of a_0 and a_i but they assume somewhat simplier forms in certain special cases. One important case is where $a_0 = -a_i = a$: this arises when the simplest case of the steering problem is considered, that is when centrifugal force and longitudinal resistances are neglected, as will be seen later. In this case equations (6), (8) and (9) become respectively,

$$R = (C + a)\left[\frac{x + 1}{x - 1}\right] \quad .. \quad .. \quad .. \quad (12)$$

$$V = \frac{r(\omega_0 + \omega_i)}{2} = \frac{r(x + 1)\omega_i}{2} \quad .. \quad .. \quad (13)$$

$$\Omega = \frac{r(\omega_0 - \omega_i)}{2(C + a)} = \frac{r(\omega_0 + \omega_i)}{2R} \quad .. \quad .. \quad (14)$$

The forces between a track and the ground

Before expressions can be developed for the resultant forces that act between a track and the ground due to the motion of the track about its instantaneous centre some assumptions must be made as to the nature of the elementary friction or adhesion forces that act between the track links and the ground and as to the weight distribution over the track.

The simplest assumptions are that the forces obey the laws of ordinary solid friction, that is that the frictional force is proportional to the normal force and acts in the

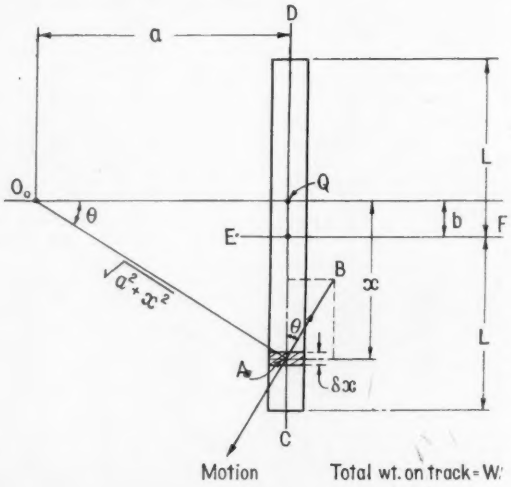


Fig. 5.

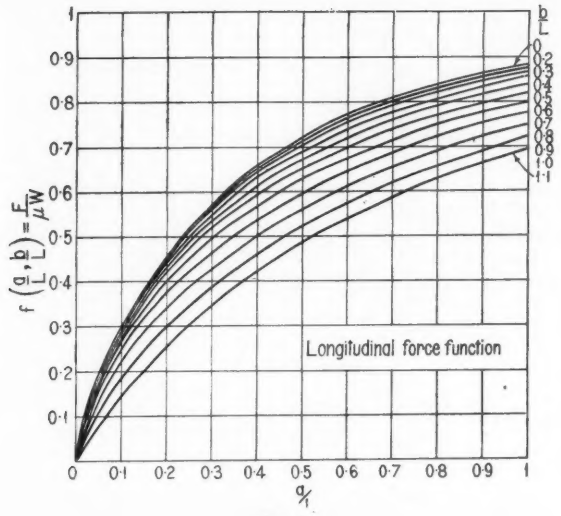


Fig. 6.

same direction as the relative velocities concerned so as to oppose relative motion, and that the weight is uniformly distributed. With these assumptions the resultant forces are obtained as follows. In order to make the expressions quite general the instantaneous centre of the track relative to the ground has been assumed to lie at a distance b in front of the transverse centre line, EF, of the track-ground contact area.

Referring to Fig. 5, if the total weight carried by the track shown is W and the length of the track-ground contact is $2L$ then the weight per unit length of track is $W/2L$ and the frictional force acting on the element of track shown shaded is $\mu W\delta x/2L$ in the direction AB. Resolving this along the centre line of the track and integrating to give the total force gives,

$$F = \frac{\mu W}{2L}\int_{(-L+b)}^{(L+b)} \cos\theta\, dx$$

$$= \frac{\mu W}{2L}\int_{-L+b}^{L+b} \frac{a\,dx}{\sqrt{a^2+x^2}} = \mu W f\left(\frac{a}{L}, \frac{b}{L}\right) \quad .. \quad (15)$$

where

$$f\left(\frac{a}{L}, \frac{b}{L}\right) = \frac{C}{2}\left[\log_e \frac{Z_2 + \sqrt{Z_2^2 + a^2}}{Z_1 + \sqrt{Z_1^2 + a^2}}\right]$$

and

$$a = a/L,\ Z_1 = (b/L) - 1,\ Z_2 = (b/L) + 1$$

For shortness the function $f(a/L, b/L)$ will generally be written as $f(a, b)$ and similarly with the other functions that appear later. Also when b is zero, as it often will be, the function will be written $f(a)$.

The longitudinal force function $f(a, b)$ is plotted in Fig. 6. for values of a/L and b/L from 0 to 1. It may be pointed out that the sign of the function changes with change of sign of a but not with that of b.

Similarly on resolving perpendicular to AB we get for the resultant lateral force,

$$P = \frac{\mu W}{2L}\int_{-L+b}^{L+b} \sin\theta dx$$

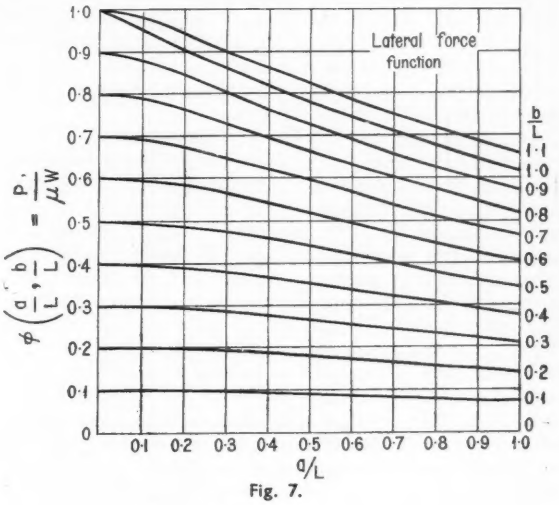


Fig. 7.

$$= \frac{\mu W}{2L}\int_{-L+b}^{L+b} \frac{x\,dx}{\sqrt{a^2+x^2}} = \mu W\phi(a, b) \quad .. \quad .. \quad (16)$$

where

$$\phi(a, b) = \tfrac{1}{2}[\sqrt{Z_2^2 + a^2} - \sqrt{Z_1^2 + a^2}]$$

The lateral force function $\phi(a, b)$ is plotted in Fig. 7. it does *not* change sign with change of sign of a but does change with change of sign of b.

Also by summing up the moments about Q of the force components perpendicular to AB the resultant moment about Q is found to be,

$$M_Q = \mu WL\psi(a, b) \quad .. \quad .. \quad .. \quad .. \quad (17)$$

where

$$\psi(a, b) = \tfrac{1}{4}[Z_2\sqrt{Z_1^2 + a^2}) - Z_1\sqrt{Z_2^2 + a^2}] - a\,.f(a, b)$$

The moment function $\psi(a, b)$ is plotted in Fig. 8. It does not change sign with change of sign of either a or b but only with change of direction of angular rotation about the centre O.

Anisotropic Friction. It is quite certain that the adhesion force for longitudinal motion of a track shoe having spuds can be very different from that for lateral motion and the two coefficients μ_L and μ_T for these

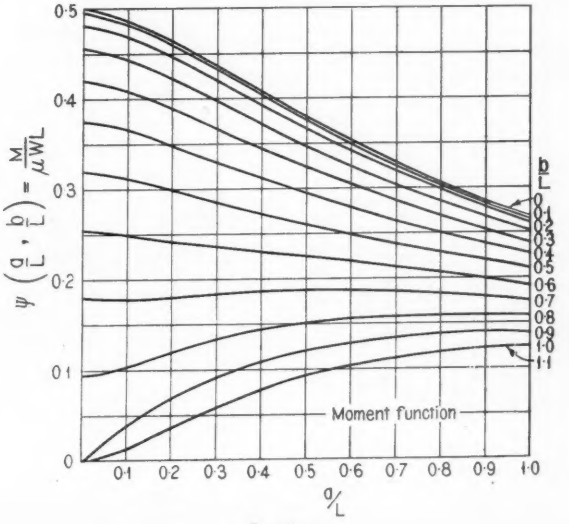


Fig. 8.

TRACKED VEHICLES (Continued)

motions can easily be found experimentally. The magnitude and direction of the adhesion force when the motion of the track shoe is in some intermediate direction are, however, quite uncertain and there is no experimental evidence available on the subject.

A suggestion by E. W. E. Micklethwait appears reasonable, however, and has the merit that it can easily be dealt with analytically; it is indicated in Fig. 9. For longitudinal motion the adhesion force is $\mu_L W$, this is represented by OA; for lateral motion the force is $\mu_T W$ and this is represented by OB, shown smaller than OA, although it might in many cases be larger. For motion of the shoe in an intermediate direction along the line OC the longitudinal component of the adhesion force is assumed to be $OE = OC \cos\theta = \mu_L W \cos\theta$ and the lateral component $OF = OD \sin\theta = \mu_T W \sin\theta$.

The resultant adhesion force on the shoe will then be OG and it is easily seen that G lies on an ellipse, shown dotted. The direction of this resultant force is different from the direction of the motion of the shoe. With this assumption the resultant longitudinal force on a *track* that is turning about an instantaneous centre is given by using the value μ_L in equation (15) and the lateral force and moment are given by equations (16) and (17) using μ_T.

Elementary dynamics of turning vehicle

The simplest possible case that can be taken is when there is no longitudinal resistance to the motion of the vehicle, the ground is level, the adhesion is isotropic, the speed and radius of turn are constant and centrifugal force is neglected. With these conditions there will be no resultant lateral force between the tracks and the ground and so it follows that b (Fig. 5) will be zero. The forces and moments are therefore as shown in Fig. 10, and it follows that,

$$\begin{aligned} F_0 + F_i &= O \\ \therefore F_0 &= -F_i \\ \therefore a_0 &= -a_i \\ \therefore M_0 &= M_i \end{aligned}$$

Hence, taking moments about G,

$$\begin{aligned} (F_0 - F_i)C &= M_0 + M_i \\ 2F_0C &= 2M_0 \\ \therefore\ 2\mu W f(a_0/L).\, C &= 2\mu W L \psi(a_0/L) \end{aligned}$$

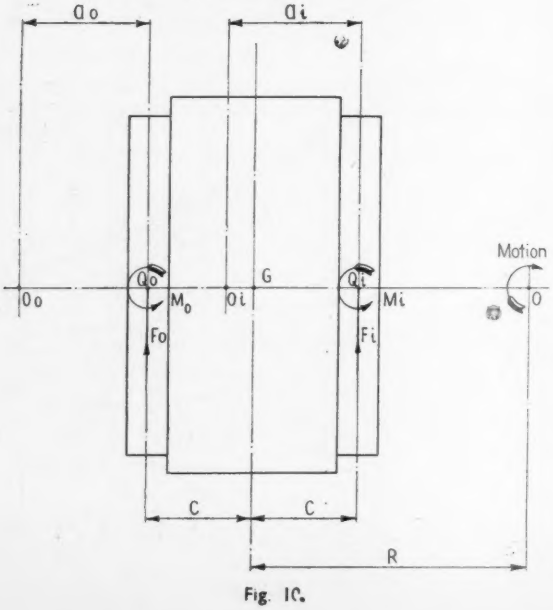


Fig. 10.

Hence $\dfrac{f(a_0/L)}{\psi(a_0/L)} = \dfrac{L}{C}$ .. .. (18)

There is only one value of a_0/L that will satisfy this equation; it may be found from Fig. 11 which is a graph of the ratio $f(a/L)/\psi(a/L)$ plotted against a/L.

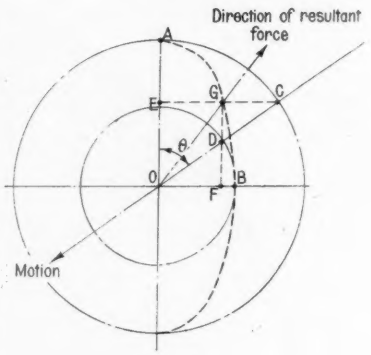


Fig. 9.

To take a practical example let $2L = 11$ft, $2C = 7{\cdot}56$ft, then $L/C = 1{\cdot}455$ and from Fig. 11 it will be found that $a_0/L = 0{\cdot}35$, so that $a_0 = 1{\cdot}925$ft. Hence from equation (12) the radius of turn is given by,

$$R = 5{\cdot}71\left[\frac{x+1}{x-1}\right]$$

and values of R for various values of x are given in the table below,

x	1·1	1·2	1·3	1·4	1·5	1·8	2·0
R (ft) ..	120	69	44	34	29	20	17

It has been found that in practice the radii of turn calculated in this manner are realized fairly closely at low speeds and on moderately hard ground.

For anisoptropic friction according to Micklethwait's assumption equation (18) becomes,

$$\frac{f(a_0/L)}{\psi(a_0/L)} = \frac{\mu_T}{\mu_L}\cdot\frac{L}{C} \quad .. \quad .. \quad .. \quad .. \quad (19)$$

Steering dynamics with Longitudinal Resistance

A slightly more complex case arises if longitudinal resistance is taken into account. Let this resistance be D, then the equations of equilibrium are,

$$F_0 + F_i = D$$

$$\therefore f(a_0/L) + f(a_i/L) = \frac{D}{\mu W} \quad .. \quad .. \quad .. \quad (20)$$

and $(F_0 - F_i)C = M_0 + M_i$

$$\therefore f(a_0/L) - f(a_i/L) = \frac{L}{C}\left[\psi(a_0/L) + \psi(a_i/L)\right] \quad (21)$$

Equations (20) and (21) determine the values of a_0 and a_i, but can be solved only by trial and error; thus guessing a value of a_0 the value of $f(a_0/L)$ is found from Fig. 6,

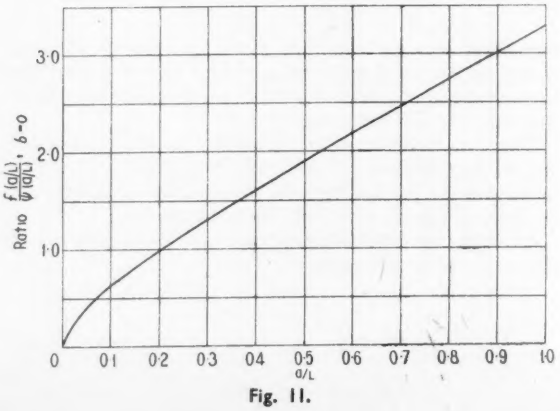


Fig. 11.

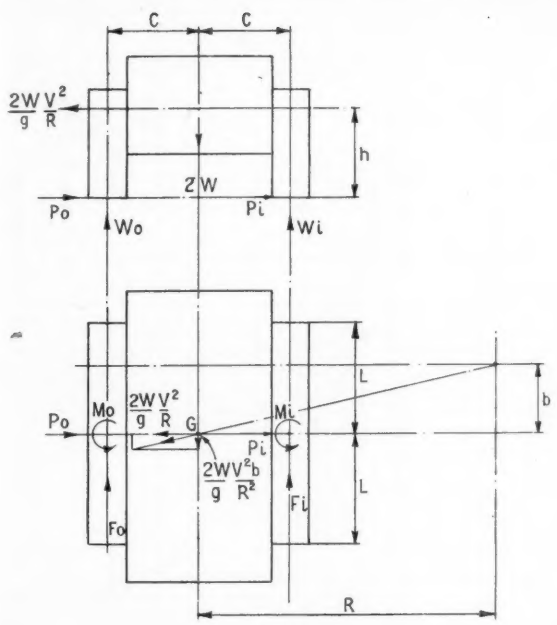


Fig. 12.

and hence the value of $f(a_i/L)$ from equation (20). The corresponding value of a_i is obtained from Fig. 6. The values of $\psi(a_0/L)$ and $\psi(a_i/L)$ corresponding to these values of a_0 and a_i are then found from Fig. 8. If these values do not satisfy equation (21), as is most probable, other guesses must be made until satisfaction is obtained. Usually only three or four trials are necessary. As before if Micklethwait's assumption is made, then the value of μ in equation (20) must be μ_L and the right-hand side of equation (21) must be multiplied by μ_T/μ_L.

Steering dynamics, more general case

In the case now to be considered the only assumptions made, in addition to those upon which the forces between the track and the ground are calculated are that the speed and radius of turn are constant. The forces and moments are then as shown in Fig. 12, and the following equations may be written down,

$$W_0 + W_i = 2W \quad .. \quad .. \quad .. \quad .... \quad (22)$$

$$P_0 + P_i = \frac{2WV^2}{gR} \quad .. \quad .. \quad .. \quad .. \quad (23)$$

$$F_0 + F_i = \frac{D + 2WV^2 b}{gR^2} \quad .. \quad .. \quad .. \quad (24)$$

$$(W_0 - W_i)C = \frac{2WV^2 h}{gR} \quad .. \quad .. \quad .. \quad (25)$$

$$(F_0 - F_i)C = M_0 + M_i \quad .. \quad .. \quad .. \quad (26)$$

In addition the following equations also apply,

$$F_0 = \mu_L W_0 f(a_0 b) \quad .. \quad .. \quad .. \quad .. \quad (27)$$

$$F_i = \mu_L W_i f(a_i b) \quad .. \quad .. \quad .. \quad .. \quad (28)$$

$$P_0 = \mu_T W_0 \phi(a_0 b) \quad .. \quad .. \quad .. \quad .. \quad (29)$$

$$P_i = \mu_T W_i \phi(a_i b) \quad .. \quad .. \quad .. \quad .. \quad (30)$$

$$M_0 = \mu_T W_0 L \psi(a_0 b) \quad .. \quad .. \quad .. \quad .. \quad (31)$$

$$M_i = \mu_T W_i L \psi(a_i b) \quad .. \quad .. \quad .. \quad .. \quad (32)$$

In these eleven equations there are sixteen quantities that can be regarded as variables and so five of them must be settled arbitrarily, or by considerations other than the dynamics of the problem, before the equations can be solved. When this has been done the eleven equations can be solved for the remaining quantities, but the process is one of trial and error and is tedious. During the 1939-45 war Junior Commander M. Knox, M.Sc., at that time a member of the author's staff at the School of Tank Technology, undertook the task of solving the equations for a large number of values of the variables. The quantities that were chosen arbitrarily were V, R, D, μ_L and μ_T.

Space is not available to give all the results that were obtained, but some of the more important will be given. Fig. (13) gives the variation of the longitudinal forces with the radius of turn for five values of the speed. It will be seen that at low speeds the forces remain constant as the radius of turn changes from 10ft up to infinity, but that for radii less than 10ft there is a rise and then a fall in the magnitude of the outer track force while the negative value of the inner track force decreases and the force becomes positive. At the higher speeds the increase in the outer track force when the radius of turn becomes small does not occur and the force decreases continuously with decrease of radius.

It will also be seen that at low values of the radius there may be two solutions of the equations and the track forces can have one of two values for the same radius. It is not suggested that these latter results are of any very great practical importance since turns at relatively high speed and small radii are not usually made. However the action of centrifugal force, to which these results are attributable, can be simulated by the sideways component of the vehicle weight when operating on a side slope and the results shown would then be obtained at lower and practicable speeds.

Fig. 14 shows the effect of the track centre distance; the results are quite as would be expected but bring out the importance of making the dimension as great as is

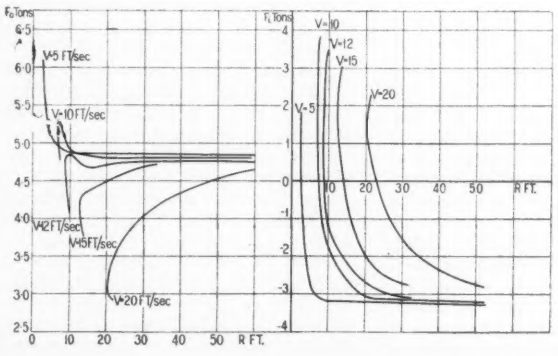


Fig. 13.

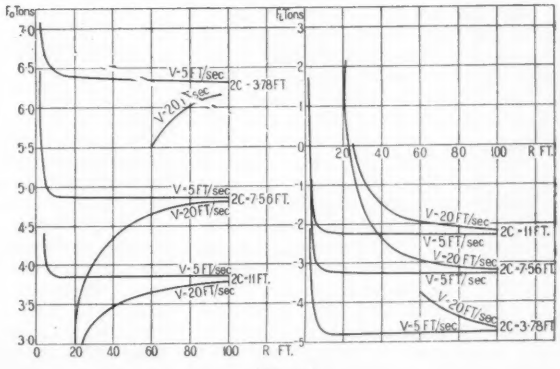


Fig. 14.

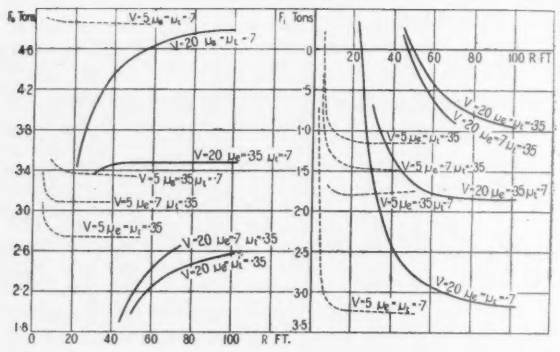


Fig. 15.

practicable. Fig. 15 shows the effect of assuming different coefficients of adhesion for longitudinal and lateral motion, according to Micklethwait's method, and indicates clearly that low values of the lateral coefficient are extremely effective in reducing the longitudinal forces necessary when turning. This implies that the spuds on track shoes should be designed to give high values to the longitudinal coefficient of adhesion and as low values to the lateral coefficient as are practicable bearing in mind the necessity to operate on side slopes.

Finally, Fig. 16 shows the variation with the radius of turn of the sprocket speed-ratio and it will be seen that for all speeds the radius of turn is closely related to the sprocket speed-ratio until the lower limit of the practicable radius is reached when, as is shown by the curve for a speed of 20 ft/sec, very large changes in the speed-ratio occur for very small changes in radius and the general trend of the curves, namely that increase in speed-ratio is associated with decrease of radius, is reversed.

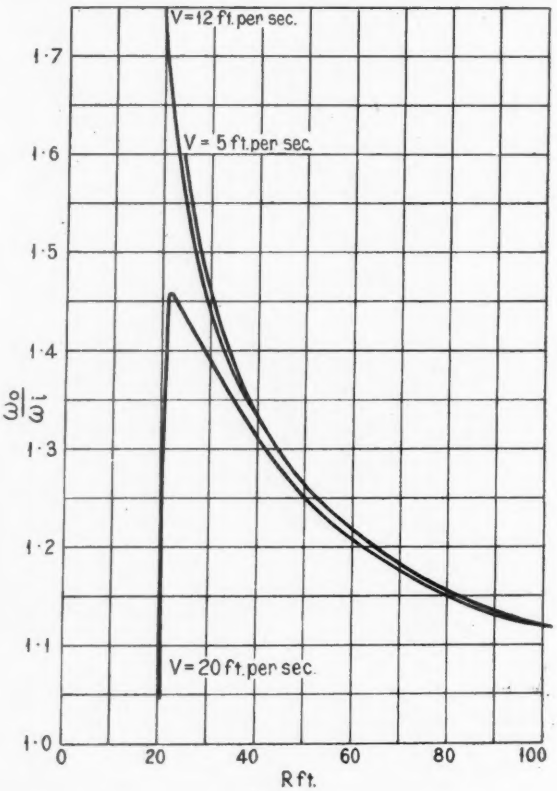


Fig. 16.

For all the above curves the following data apply, except where otherwise stated on individual graphs. $2W = 19$ tons, $2L = 11$ft, $2C = 7{\cdot}56$ft, $D = 1{\cdot}59$ tons, $h = 4$ft, $\mu = 0{\cdot}7$, $r = 1{\cdot}058$ft.

(To be continued)

RESETTING DIAMOND TOOLS

A "CONTRACT" resetting service for diamond tools has been introduced by Engineering Diamonds, Ltd., 26, Warwick Row, Coventry. Under this service the resetting of all single natural diamond tools used for wheel dressing is undertaken for a period of twelve months at a fixed monthly charge, based on the average quantity involved.

This scheme has been introduced particularly for the larger users of diamond tools. The monthly charge is on a sliding scale with 41 tools per month as the minimum. The greater the number of tools, the greater the overall saving. In addition to the tangible saving the system also has the following advantages:—

(1) The frequency of resetting can be fixed to obtain the best possible diamond life.

(2) Clerical work is reduced, since only one initial authority and one invoice per month are required instead of many separate orders.

(3) Every tool is marked with the weight of diamond on resetting. This allows the relative performance of any diamond to be determined.

Tools do not remain in the works more than eight working hours and normally they are returned on the day they are received. Whether "contract" or ordinary resetting is undertaken, the fullest co-operation is given where the user wishes to maintain any system of ensuring that as the diamonds get smaller they are used on lighter work. (1871)

INSTITUTION OF METALLURGISTS

THE next examinations for the Licentiateship and Associateship of the Institution of Metallurgists will be held from August 28th to September 4th, and candidates must submit their applications for permission to enter the examinations before June 1st. Each application must be made on a form obtainable from the Registrar-Secretary, The Institution of Metallurgists, 4, Grosvenor Gardens, London, S.W.1, and must be accompanied by a registration fee of one guinea.

ENGINEERING TRAINING

ONE of the more encouraging trends in industry to-day is the greatly increased interest shown by many organizations in the selection and training of young entrants. This will go far to ensure an adequate supply of technicians and craftsmen in the future. A well thought out training scheme for boys is described in a booklet recently issued by The Rootes Group Manufacturing Division. It makes clear that every care is taken to ensure that an applicant has the necessary aptitude before he is committed to an engineering career.

Successful applicants enter into either pupilage or apprenticeship. Pupilage is for boys starting between the ages of 17 and 18 and who have reached matriculation standard. Apprenticeship is for boys starting at 16 years. Carefully planned technical and practical training is given to both groups.

Primarily, this booklet is issued for the information of parents. It will also be of interest to organizations considering the introduction of a training scheme and to those already having such schemes.

DEVELOPMENT-RESEARCH

An Addition to the Engineering Department of Humber, Ltd., Extending the Range of Experimental Work

IN a paper presented to the Institution of Automobile Engineers some years ago by A. G. Booth, an account was given of the Experimental Department of Humber, Ltd. Recently A. Craig Miller, M.I.Mech.E., M.Inst.Pet., head of the Development-Research Department, Humber, Ltd., Coventry, gave some further details of the Company's experimental establishment in a lecture entitled "Research as Applied to the Motor Industry," presented to the Belfast Association of Engineers. The following abstract gives some account of the latest developments.

Some six years ago Humber, Ltd., decided to form a Development-Research Department as an entirely separate concern from the main Engineering Department, with the object of carrying out a fairly wide range of research and development outside the normal development carried out by the Experimental Department. As the name Development-Research suggests, the work was not intended to be so much pure research as a mixture of some research and some long-term development not intended for immediate production, in order to place at the disposal of the management information and tests on new devices so that they should be in a position to assess their value or desirability for incorporation in the product.

A two-storey building

The department is housed in a two-storied building remote from the rest of the factories, in order that a certain amount of secrecy may be maintained. The ground floor has an area of about 4,000 sq ft containing the machine section, test section, etc., and an upper floor of about 3,000 sq ft containing offices.

The equipment at present carried in the department is as follows:—

- 10ft by 5ft slotted bed plate.
- 12ft by 3ft slotted bed plate.
- Avery 20-ton pulsator fatigue test machine.
- Parkson universal milling machine.
- South Bend 6½in centre lathe.
- Le Blond 10in centre lathe.
- Brown & Sharpe No. 2 universal grinder.
- Asquith O.D.I. radial drill.
- Sheet metal rolls.
- Laycock car hoist.
- Amsler 20-ton universal test machine.
- A complete range of electric motors from 15 h.p. to 1/10 h.p.
- A Crompton-Parkinson variable speed 10 h.p. motor and M.G. set.
- Heenan & Froude eddy current dynamometer.
- Three cathode ray oscilloscopes.
- Stroboflash and equipment.

It will be evident that the department is able to manufacture almost anything likely to be required for development. Since its inception the departmental staff has been maintained at about twenty, always including a high percentage of qualified engineers.

One of the important functions of the department is the design and manufacture of apparatus capable of producing accelerated tests of components, etc. Some of these test rigs have led to highly satisfactory results, while others have provided results that are interesting but not of great practical value.

Friction lining research

During the development of magnetically operated friction clutches, it became necessary to find a clutch friction material that would have a much better resistance to fade than the kind of lining which was proving satisfactory with a conventional foot-operated clutch. Counters were fitted to the clutch pedals of a number of cars undergoing different types of test, and the clutch usage varied between 4,100 operations in 1,100 miles down to an average of 25 operations per mile in London traffic. This information indicated that with an automatically operated clutch having considerably more slip than a foot-operated clutch, the fade characteristic of the lining was of paramount importance. The test rig was therefore made up to produce fade characteristics rapidly. A 15 h.p. motor was coupled to an engine from which all the reciprocating parts had been removed. A vertical distributor was connected to a gear box having a reduction of 700 to 1, which resulted in a speed on the output of the gear box of 1 r.p.m.

Cycle of operations

An arrangement was therefore added that would result in a cycle of operations of one per minute. On the face of the gear box are a number of studs with which a charging arm fixed to the output shaft progressively makes contact. The studs are connected to the resistances in such a manner that the voltage applied to the magnetic clutch was increased progressively. For test purposes the voltage was applied much more slowly than would happen in practice. This was done to increase the slip and therefore the tendency to slip and fade. The clutch was located in a normal manner on the crankshaft and coupled through the normal transmission to a very large filled flywheel. This flywheel was equipped with a band brake applied by a compressed air servo motor controlled by a solenoid.

The cycle of operations began by engaging the clutch on the studs so that it was made to accelerate the flywheel from rest. By using different gears different speeds were obtained, and therefore different mean torques to accelerate the flywheel. As soon as steady conditions were reached, the wiping arm made contact with a further stud that actuated the band brake solenoid after the clutch had been disconnected and so brought the flywheel to rest in time for the beginning of the next cycle. This test rig was in use for more than twelve months, and with the collaboration of the lining manufacturers, a very great improvement in fade characteristics was obtained.

Universal joints

Another interesting test rig was evolved in the attempt to produce scuffing and seizure of the "Tracta" type universal joints as used on F.W.D. military vehicles. It was decided to use a variant of the Gleason "four-square" axle test, and to this end four front differential assemblies were arranged in the form of a "close" circle having their differentials permanently locked. In one leg of the circle a transfer gear box was inserted, driven by an electric motor, while on the opposite leg a torsion coupling was inserted. It will be seen that by loading the system with the torsion coupling, any desired torque would be transmitted in turn by all the joints without the necessity for providing a large amount of power to drive the rig. Various tests were carried out with torques between 13,000 and 16,000 lb/in.

Generally speaking, the approach of the motor trade to the question of noise and vibration has not been a scientific one, and the Department has for a long time been hoping to be able to make a real study of this very important subject. To this end a set of vibration testing equipment has been designed and made. It comprises a resistance-coupled oscillator having a frequency range of 5 to 500 cycles per second and carefully designed to eliminate harmonics, etc. The oscillator feeds a 50 watt amplifier which in turn supplies the vibrator. The vibrator consists of what is in effect a large moving coil loud speaker without any cone, but with provision for the

DEVELOPMENT-RESEARCH (Continued)

moving coil to be connected to any other object by a rigid rod.

The equipment can be used either to determine the frequency response of a complete vehicle, or more particularly to locate the resonant points and determine their resonant frequencies. This is done by coupling the vibrator to various parts of the complete vehicle, such as the floor, the chassis frame or a bumper bracket, and to go through the frequency range noting both aurally and with a vibration pick-up and C.R.O. which panels, etc., are resonant and at what frequency.

Resonance exploration

The compilation of a complete resonance exploration of a car is a fairly lengthy procedure and we have not had this apparatus working long enough to complete an investigation, but the future programme includes the complete resonance testing of all vehicles produced by the Companies which this Department serves, and it is hoped and expected that a considerable improvement in general noise and vibration will be achieved.

A further use of the equipment is the fatigue testing of components at fairly high frequencies which may be outside the scope of mechanical testing. The oscillator may be used to produce the frequency in question or alternatively a strain gauge may be attached to the component at the point in question or a microphone near it connected through a suitable phasing circuit to the input of the amplifier back to the vibrator, which thus maintains the oscillation at the resonant frequency. In this way large fatigue stresses may be initiated with a low-powered unit. Our initial work on this subject, however, indicates that a 50-watt amplifier is not powerful enough to cover our requirements and more powerful equipment is in the course of being designed.

Engine test correlation

The correlation of engine test results on the bench and road performance has been a matter over which much thought has been spent. Various formulæ for determining the road resistance, etc., of vehicles have been produced and used with a certain amount of success, but it has always been felt that a means of actually measuring the torque or power which was being transmitted from the engine would be of great value. At the moment we are constructing a device for measuring the torque output from the gear box and preliminary tests indicate that this should be successful, although we do not know as yet what degree of accuracy may be obtained. The apparatus consists of a tubular casing which is in effect an extension of the gear box towards the rear axle. Inside this casing, mounted on heavy-duty ball bearings, is a steel shaft of suitable diameter to produce torsional deflection when subject to the kind of torque that is likely to be measured. At each end of this shaft are mounted sine wave generating discs, which, in conjunction with a permanent magnet and pick-up coil, provide a sine wave, the magnitude of which is proportional to the speed of rotation. The two discs have their pick-up coils connected in opposition, so that at any speed of rotation they cancel each other out. When, however, the shaft is transmitting power, a small angle of twist is produced in it, which presents itself electrically as a phase shift in one of the sine waves equal to the angle of twist times the number of poles on the disc. Under these conditions the two waves do not cancel out, but there is a resultant proportional to the phase shift and thus to the torque. In this way an output is obtained which varies as the speed times the torque, and by suitably calibrating a voltmeter, direct readings of power can be taken without absorbing any power in the measuring equipment and without the use of slip rings or other dubious connections.

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

Accumulator Charging, Maintenance, and Repair

By W. S. Ibbetson, B.Sc., A.M.I.E.E., M.I.Mar.E.

London: SIR ISAAC PITMAN & SONS, LTD., Parker Street, Kingsway, W.C.2. 1949. $4\frac{3}{4} \times 7\frac{1}{4}$. 193 pp. Price 10s.

Accumulators are now employed for a wide variety of purposes, such as emergency lighting, miners' electric lamps, power for auxiliary plant, etc. It may have been thought some years ago, when the network of power distribution from large generating stations began to cover the country, that the use of the accumulator would soon be confined to motor cars and radio sets in remote country districts. The fact that this is far from the case is proved by the fact that ten editions of this little book have been necessary since 1928. This, the tenth edition, is a complete revision and includes a good deal of new material.

It is a practical book for the user and maintainer of accumulators. The reader need know nothing of electricity or of mathematics to understand all the text, even the introductory chapter dealing with circuits, volts, amperes, Ohm's Law, e.m.f., etc.

Over fifty pages are devoted to charging batteries on d.c. supply circuits and this includes the charging of batteries at home, charging boards, earthed circuits, the electrolyte, first charge, h.t. batteries, trickle charging, generator charging, electric vehicle batteries, etc. Charging from a.c. circuits includes descriptions of rectifiers and convertors. A valuable chapter deals with battery diseases and their treatment and includes sulphating, de-sulphating agents, buckling, shorts, sludge, temperature troubles, etc. Notes on repairs to batteries will be extremely useful to workshop personnel and maintenance engineers. The two types of alkaline accumulators, the nickel-iron or Edison cell and the nickel-cadmium or Jungner type are dealt with in sufficient detail for the practical user and methods of storing these batteries are included.

The section covering country house, private and emergency sets includes the latest (1946) L.C.C. regulations relating to the use of emergency batteries in cinemas and adequate descriptions of the types of plant employed for this purpose are provided. Three appendices complete the book. The first is concerned with the cadmium test. The second is the syllabus covering batteries for the motor vehicle electrician's course for the City and Guilds of London Examinations and a number of typical questions set at these examinations are given, some having worked solutions. The third appendix deals with the care and maintenance of miners' electric lamps. The need for technical training in lamproom maintenance has been recognized by the Ministry of Fuel and Power and a staff college has been established at Nuneaton for the instruction of head colliery-lampmen and those in charge of lamp installations at mines.

Analysis and Lubrication of Bearings

By Milton C. Shaw and E. Fred Macks.

LONDON: MCGRAW-HILL PUBLISHING CO., LTD., Aldwych House, W.C.2. Published in the U.S.A. (Price $10.00). 1949. 6×9. 618 pp. Price 85s.

If it were possible to lubricate a bearing, of any type, size or material, so that the shaft was at all times fully floating and metallic contact never occurred under any conditions of loading, the bearing and shaft would obviously outlast the useful life of the machine. As a result, production would be considerably increased and maintenance costs substantially reduced.

Many machines are still manufactured today with little or no thought for lubrication, a mere oil hole with no dust cap, or a difficult-to-get-at greaser are frequently the extent of the designers efforts concerning lubrication. Happily, this state of affairs appears to be changing due to the keener competition between manufacturers and the increased interest, and education, of the buyer who is now paying greater attention to maintenance costs. This volume should therefore be studied by all connected with bearing design, and they must not expect to assimilate all the knowledge contained therein at one reading.

The authors state "The purpose of this book is to present the fundamental concepts underlying the design and perform-

ance of shaft bearings." The book does more than this and it should become the standard reference work on bearing design and lubrication, in as far as these two subjects are combined. It only includes infrequent notes concerning metallurgy, and it does not include practical details of circulation systems for bearings or lubricators, but it is doubtful if there is any published work containing such a unified exposition of the many aspects of bearing analysis.

The reader should have a working knowledge of applied mechanics and mathematics and should be familiar with the calculus. This is a book for advanced students but is also intended for the engineer in industry. It serves to train the student in the analytical approach to general engineering problems. "Tricks-of-the-trade" and empirical formulæ are criticized and, in many cases, logical reasoning points to the more satisfactory methods of solution.

Early chapters are devoted to the general principles of bearing load analysis and dimensional analysis respectively. This includes generalized load charts and polar diagrams for crankpin loads. A series of chapters is then devoted to hydrodynamic lubrication including journal and thrust bearings and high speed considerations. Technical and practical considerations regarding oil holes and grooves, hydrosphere spherical bearings, air as a lubricant, fluctuating loads, etc., are some of the subjects dealt with. High speed considerations cover turbulence in the oil film, oil-film whirl, journal mass, inherently stable bearings, shaft whirl, dry friction whirl, bearing erosion and its effects, cavitation, etc., etc. Anti-friction bearings are concisely dealt with in fifty pages and include details of the work of Stribeck and Styri, Palmgren's equation and notes upon such diverse matters as elastic deformation and gyroscopically induced slip.

Realizing that boundary lubrication needs an empirical approach after consideration of basic principles, the authors have not delved too deeply into the complex subject of extreme pressure additives and chemicals, but sufficient information is provided to form an excellent basis for further study. As elsewhere throughout the book, numerous references to published literature are given so that the reader can pursue any branch of the subject further if desired. To complete the book, a number of representative bearing testing machines and their characteristics are discussed. Some of these are better known in the U.S.A. than in this country. They include the Morgan and Muskat machine, Nutcracher type of machine, cageless roller-bearing tester, four-bearing friction machine, DVL, BTH and Westinghouse testing machines, etc.

The New Foremanship. Volume Two

By F. J. Burns Morton, F.I.I.A., A.M.I.P.E.

London: CHAPMAN & HALL, LTD., 37, Essex Street, W.C.2. 1949. $5\frac{1}{2} \times 8\frac{1}{2}$. 187 pp. Price 12s 6d.

It would seem as if the object of this book is to make out a case for the formation of an Institute of Foremen. The author has collected his data from home and overseas and the majority of it would support such an objective. Whether this is the primary aim or not, the need for such a professional body is certainly emphasized. When it is realized that every type of manufacturing industry has its foremen and has employed them to act, generally, as a link between management and men, for generations, and that there are probably three quarters of a million foremen in this country, the value of such an institute is at once apparent. The author is aware of the difficulties of forming such an institute and of the time needed to achieve its full status. The fact that every type of differing industry would be represented also presents a problem which is not discussed. The fundamental principles of foremanship and the training of foremen for most industries are, however, matters which require increasing attention and such work is rightly the responsibility of such a body.

The author speaks from experience. Volume One of this work put forward a scheme for training within industry, and this scheme has since been adopted by the Ministry of Labour and National Service. The scheme has been tested and proved sound. As a result of experience gained in this connection, the author has presented an impartial examination of this scheme in some sections of this book.

Information on training for foremanship and progress made in these directions in other countries is given and the facts show that several countries, especially the U.S.A. and Australia are ahead of Great Britain in their recognition of the value of correct education and status for foremen. University Colleges have provided courses for a number of years with obviously successful results. In Great Britain, early work was done by several leading industrial concerns for their own staffs, and National action was not taken until 1941 when the Ministry of Labour and National Service instituted courses at technical schools all over the country and which were attended by 20,000 foremen. It is interesting to note that the average age of 1,400 foremen who attended the course organized by the Cotton Control Board Recruitment and Training Department was 44 years. The need is for training from an early age so that foremen can be selected from those who show aptitude and have had correct practical and "technical" training.

The importance of the foreman in the establishment of the new conditions which are evolving in industry and his place in industry afterwards provide the keynote behind this book. Sir George Beharrel states in his foreword: "his ability to face the new conditions may well be the measure of the success of industry." With this in mind, the author has discussed such matters as the relations between managers and foremen, works morale, managing men, managing women labour, foremen and trade unions, the use of psychological methods, and other problems.

Throughout the book the author shows a realization of the fact that present methods of supervision are not perfectly satisfactory, and this state of affairs has been caused by several contributing factors. Foremen, when appointed, frequently hold their positions for thirty years so that training for older and experienced foremen should be introduced. Suitable methods of recruitment and selection of personnel for training as foremen have not up to now been available. It is rightly proposed that the recruitment of foremen should not be confined to the workshop, "men must be sought with a broad general education, a fair standard of intelligence, a good capacity for learning quickly, and ability for rapid adjustment to changing circumstances." The future foreman must be treated as the manager of his department and must be capable of convincing his management that his influence over employees through supervisory leadership will pay dividends.

Extraction and Fabrication of Aluminium

THE NORTHERN ALUMINIUM CO., LTD., Banbury, Oxfordshire. 1949. 6×9. 115 pp.

The Northern Aluminium Company are to be congratulated in producing this book, not only because of the interesting and valuable information that it contains, but also because of its high quality as a book production. Obviously much time and thought has been devoted to its design and layout and credit is due to Editorial Consultants, Ltd., who were responsible. Such a book must add to the prestige of the company.

Few people have any detailed knowledge of the production of aluminium. The industry has grown relatively quickly. One hundred years ago this ubiquitous material was almost a precious metal. Now its industrial uses are second only to those of steel. It is doubtful if any complete treatise on the production, extraction and fabrication of aluminium exists. This is surprising in view of the importance of aluminium in modern industry and is still more surprising in view of the interesting history and modern processes involved. This book gives the story up to the point when the metal leaves the manufacturer for conversion into finished products by further processing.

The book is divided into three main parts, but there is a fourth section briefly setting out and illustrating the uses of aluminium and its alloys. The first section deals with the extraction, beginning with its historical associations, and covering the sources of aluminium ores and bauxite, the mining and preparation of alumina, and details of the world's production of aluminium and bauxite.

Part two covers the properties and part three the fabrication of aluminium and its alloys. This is the longest section of the book and deals with remelting, pouring, cast structures, rolling sheet, hot rolling, annealing, cold rolling, finishing, extrusion, presses, heat treatment, drawing, forging, sand casting, die-casting, etc., etc.

Although written primarily for readers with technical knowledge, the book should appeal to anyone of mechanical bent. It is the very many half-tone illustrations that provide this general interest. All the photographs of rolling and extrusion, and many others, were taken in the Banbury and Rogerstone works of the Northern Aluminium Company, and those of casting and forging in their Birmingham works.

These well-taken and well-reproduced photographs tell the story themselves. It is well that those who use aluminium in quantity, as indeed do all the automobile industries, should have a clear picture of the processes involved in its manufacture. Whether it be technical information concerning what can be done with aluminium after manufacture, or intricate details concerning its processing, this can be obtained from various technical papers and text books. An insight into the manufacturing methods, the machines and processes employed is complementary to the technical detail, and it is in this particular connection that this book is valuable.

Welding Dictionary

By R. N. Thompson and G. Haim.
London: THE LOUIS CASSIER CO., LTD., Dorset House, Stamford Street, S.E.1. 1950. $7\frac{1}{4} \times 4\frac{3}{4}$ Price 21s.

Engineers who are sufficiently interested in welding, or those engaged in this industry, frequently wish to study the articles and papers on the subject printed in foreign journals and magazines. Although they may be able to read the languages, they inevitably find difficulty in translating the many technical expressions, words and phrases peculiar to a specialized industry. This book gives translations in French, German, Spanish and, of course, English, of every word used in connection with welding. For example, such expressions as weld bead, flashing-time, fish eyes, convex filler weld, double-flanged butt weld, erratic arc, micaceous iron oxide, width of kerf, stress relief heat treatment, rightward welding, pre-heating torch, etc., are given. Where corresponding terms do not exist for a particular word in all languages, a brief definition is given. Where one word can be translated in two or three different ways, these alternatives are included, possibly with slightly different shades of meaning, but the original word is accompanied by more than one reference in its appropriate glossary.

In addition to terms specifically related to welding, many indirectly related ones concerning electrical engineering, metallurgy and correlated subjects have been included. All electrical terms relating to electrical welding equipment will be found, and the dictionary is not by any means confined to single words.

It should fulfil its object admirably, by enabling anyone with knowledge of French, German or Spanish, to read and fully understand any technical paper concerned with welding.

The History of Engineering in Ipswich

By The Ipswich Engineering Society.
Ipswich: W. S. COWELL, LTD., Butter Market, Ipswich. 1949. $6\frac{1}{2} \times 9$. 135 pp. Price 10s 6d.

To celebrate their Jubilee 1899-1949, Ipswich Engineering Society have issued an interesting book covering the history of engineering activities in Ipswich and district. At the present day, 10,000 people are engaged in engineering work in those parts, this very considerable industrial foundation being laid by Robert Ransome. With a single workman, Robert Ransome, in 1781, started a foundry in Ipswich town for the casting of ploughshares, and this enterprise constituted in fact, the foundation of the not inconsiderable present day engineering industry there.

The Society's book opens with a brief account of the foundation of the Society. An account is also included of its founders and officers. Among them may be recognised many names now quite well known in the larger engineering world generally. The chapter that follows, covers the history of the Port of Ipswich and its development. Later chapters are devoted to descriptions of the local services, gas undertakings, transport, etc. These chapters are followed by sections dealing with the foundation, history and work of the leading local engineering concerns.

Amongst the many organizations now established in the neighbourhood is a branch of a well-known company in the United States, and these local concerns cover a wide range of engineering products, electrical, constructional, and general, both large and small. The book can confidently be commended to all engineers interested in technical histories, more particularly, of course, those who have local associations. These brief specialised accounts, although dealing chiefly with local activities, quite definitely contribute to the history of this country and are to be valued accordingly.

Private Companies: Their Management and Statutory Obligations

By Stanley Borrie.
London: JORDAN & SONS, LTD., Chancery Lane, W.C.2. 1949. $5\frac{1}{2} \times 8\frac{1}{2}$. 220 pp. Price 8s 6d.

The last edition of this work was published in 1946, and the present one is a revision to cover the provisions of the Companies Act, 1948. It therefore constitutes a standard work for directors and company secretaries concerning company law, its restrictions and privileges, for private companies. Notes are included in some instances to show the differences between private and public companies.

Company law is a subject that must be known to all secretaries, although none of these, except those acting for the largest concerns, can be expected to have studied the legal side of their business very deeply. For this reason this book must be valuable. It includes all the basic knowledge that is required since, usually, if more detailed information is sought, the secretary will invoke legal advice. The information in this book should, if not known, be available for reference.

Sections deal with restrictions and privileges, incorporation, agreements, members and shares, capital, dividends, directors, loans and debentures, meetings, annual returns, etc., etc. A list of documents required to be lodged with the Registrar, giving name of document, who signs it, dates to be lodged and penalty for default, are included. There is also a most valuable index for quick reference.

Mechanical World Electrical Year Book

Manchester: EMMOTT & CO., LTD., 31, King Street West, Manchester, 3. 1950. $4 \times 6\frac{1}{4}$. 360 pp. text and 58 pp manufacturers' announcements. Price 3s.

This little book is really too well known to need review, being now in its forty-third year of publication. It is doubtful if there is any electrician who does not know of it or has not made use of it at some time or another. This edition includes a new section on electrical welding and covers equipment, applications, arc stabilization, multiple-operator sets, power factor, D.C. welding plant, cables and electrodes, arc cutting, atomic-hydrogen arc welding, spot welding, constant timing devices, projection welding, spot welding, constant timing devices, projection welding, seam and butt welding etc.

The section dealing with meters and measuring instruments has also been extensively revised and now includes modern methods with descriptions of their principles of operation. Instruments described include hot wire, thermocouple, dynamometer, moving iron, induction, electrostatic, rectifier, oscillographs, graphic, etc. Apart from minor changes, other sections remain as before and provide a handy reference for day-to-day queries.

CORRESPONDENCE

Correspondence on subjects of technical interest is invited. The name and address of the writer must be given, though not necessarily for publication. No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter. If a reply by post be desired, a stamped addressed envelope should be enclosed.

EPICYCLIC GEARS

SIR,—Referring to my simplified formulae for epicyclic gears published in the *Automobile Engineer* for February 1950, p.73, may I point out three errors?

1. Figs. 3 and 5 have unfortunately been transposed in relation to their inscriptions.

2. The final formula for Fig. 5 is:

$$n_b = n_a \cdot \frac{a}{b} + N. \frac{a+b}{b}$$

3. The final formula for Fig. 7 is:

$$n_a \cdot \frac{(a+d)a' + d'a}{(a+d)(a'+d')} = n_c$$

You have commented in the text that the tabular method has the advantage of giving the revolutions of the planet pinions on their studs. The following example will show that this can also be found using my method:

By simple modification the formula of Fig. 5 can be written

$$n_b = (n_a + N)\frac{a}{b} + N$$

With $n_a = 1000$ r.p.m., $N = 500$ r.p.m., $\frac{a}{b} = \frac{24}{60} = 0.4$ and $b = 2c + a = 60$
$= 2 \times 18 + 24$ so that $c = 18$, we have

$$n_b = (1000 + 500) \times 0.4 + 500$$
$$= 1100$$

Since the speed of the planet pinion

$$n_c = (n_b - N)\frac{b}{c}$$

we have $n_c = (1100 - 500)\frac{60}{18}$

$$= 2000 \text{ r.p.m.}$$

ARTHUR BALOGH
Budapest.

Institution of Mechanical Engineers, Automobile Division

PETROL INJECTION

An Application to Automobile Engines

By R. Barrington, M.Sc., A.M.I.Mech.E.* and E. W. Downing*

The paper recounts the results obtained in a series of experiments carried out with the object of examining the claims made for improved results, both in fuel consumption and maximum power, which could be obtained by substituting a petrol injection system for a carburettor.

The experiments, carried out on four-cylinder engines of approximately 1.5 litres capacity, led to the development of a satisfactory system capable of giving equal fuel distribution to all cylinders with a tolerance of ± 2 per cent at all loads. They showed that, for engines of this order of size, manifold injection was superio to direct injection and established that petrol injection will not give any better combustion efficiency than carburation. Hence any saving in fuel consumption obtained can only be due to better fuel distribution between cylinders, better matching of fuel/air ratio to a desired value, or increase in compression ratio made possible by the elimination of the hot spot. Under certain conditions the resultant saving in fuel at part load on bench tests could amount to some 10-15 per cent.

Maximum power can be increased by some 15-20 per cent by elimination of the choke, and still more if the compression ratio is increased slightly to take advantage of the reduced temperature of the incoming charge. Road tests in general confirmed the results of bench tests, but brought out the effect of many other factors which affect fuel economy. In order to properly evaluate these a number of experiments were carried out which are of interest and value as affecting the problem of fuel economy in general.

Starting at low temperatures requires very considerable over-fuelling with a petrol injection system, just as in the case of a carburettor, but such a system may be made to give better control during the process of warming up. In the authors' view an engine fitted with petrol injection is pleasanter to drive as a result of its inherently better idling, more uniform and smoother torque and increased power at full throttle. The results obtained, however, while attractive, do not bear out some of the more exaggerated claims sometimes made by advocates of petrol injection systems. No attempt is made to urge the introduction of petrol injection and it is left to the industry to decide whether, in view of the results obtained, it is felt that a petrol injection system with its concomitant increase in first cost is worth adoption.

THREE years ago, the authors were set the task of investigating whether petrol injection was a worth-while proposition for the ordinary automobile engine. The investigation was to be impartial in character and to set out plainly the findings, whatever resulted, for the interest, and perhaps benefit, of the automobile industry.

Petrol injection was in general use on German aircraft engines during the 1939-45 war, but there appeared to be very little published information on injection work in connection with small multi-cylinder engines.

To be worth-while, the system must offer sufficient attractions in the form of improved economy or performance, and must be entirely reliable. Unless petrol injection can produce unique and desirable features of performance, the merits must be assessed upon the relation of fuel saving to first cost. While the cost can be estimated with some accuracy, the amount of fuel saved under widely varying conditions of service can only be ascertained by trial.

To this end, a petrol injection system has been evolved incorporating as many as possible of the features which seemed initially desirable and others which became apparent during the development. It has been bench tested in conjunction with a pre-war engine and has run on the road for 20,000 miles with promising reliability and improved economy. Later work on a modern post-war engine has also given sufficiently promising bench results to warrant assembly into a vehicle for road tests.

The type of fuel injection system used has obviously considerable bearing upon the engine results obtainable; that used on these tests is probably novel and requires some explanation in detail.

Development of the fuel system

General Description. The fuel system is built around the need for accurate metering and consistent atomization over the whole load- and speed-range of modern high-speed engines.

The chief elements are :—

(*a*) The constant pressure pump.
(*b*) The metering distributor.
(*c*) The atomizers.
(*d*) The mixture control.

The mixture control operates on the metering distributor and the other units are connected by pipes disposed according to the type of installation employed.

The metering distributor is interposed between the pressure pump and nozzles and connects the appropriate nozzle to the fuel pressure supply at the right time, permitting only a metered quantity of fuel to pass.

The Constant Pressure Pump. As a result of a number of bench tests, it was found that a minimum pressure of 75 lb per sq in was needed to produce good atomization in the small spray-angle nozzles developed for this work. It was therefore decided to develop the system around a pressure source of of 100 lb per sq in and a motor-driven aircraft-pump, with its associated pressure control, set to give a steady 100 lb per sq in at all flows, was used for bench testing.

The Metering Distributor. The construction is shown in Fig. 1. The unit consists of a rotating sleeve carrying two ports drilled at right-angles to one another, but displaced axially. Inside the sleeve is a free piston which shuttles, under the influence of fuel flow, between a fixed and a controllable stop. The sleeve, driven at half engine speed through a dog, rotates in a body which has four inlet ports connected through a common feed to the inlet from the pressure pump. In a plane at 90 deg. to the inlet ports, four outlet ports lead to the injectors for each cylinder.

As shown in Fig. 1*a*, fuel is entering the rear upper feed-port, flowing along the sleeve bore and displacing the free piston towards the control stop. This action discharges fuel from between the piston and the control stop, through the port in the sleeve, which is now opposite an injector port in the body, and thus to the nozzle situated in the appropriate cylinder, or manifold branch. When the free piston reaches the control stop, there is no further fuel flow, and continued rotation of the sleeve completes the cut-off to that cylinder.

Fig. 1*b* shows the position of the sleeve during injection to the next cylinder in correct sequence. The port at the front of the sleeve is now over the lower front feed-port, whilst the rear sleeve-port is opposite the next injection-port. Fuel flow along the sleeve bore is now reversed, the free piston travelling an exactly equal distance back to the fixed stop at the rear of the sleeve. This displaces an equal quantity of fuel through the next injector, and, upon reaching the stop, flow once more ceases. Rotation continues, bringing this rear sleeve-port once more over a feed-port so that reversal occurs again, with distribution to a third cylinder, and so on.

The travel of the piston between stops determines the quantity of fuel injected, and the control stop is positioned by either a hand control unit for bench testing, or the mixture control in the final arrangement.

The Injectors. Fig. 2 shows the type of injector used throughout this work. They are required to give good atomization at 100 lb per sq in and to function as perfect stop-valves during periods between injections. Preliminary tests had shown that, in order to prevent troubles from particles of dirt, fuel oxidation products, etc., it was better to ensure that the poppet valve in operation vibrated at high frequency clear of its seat, before finally closing. Under those conditions, the valve never appeared to stick in the open position, thus spoiling subsequent injection.

* Research Engineer, Joseph Lucas, Ltd.

PETROL INJECTION (Continued)

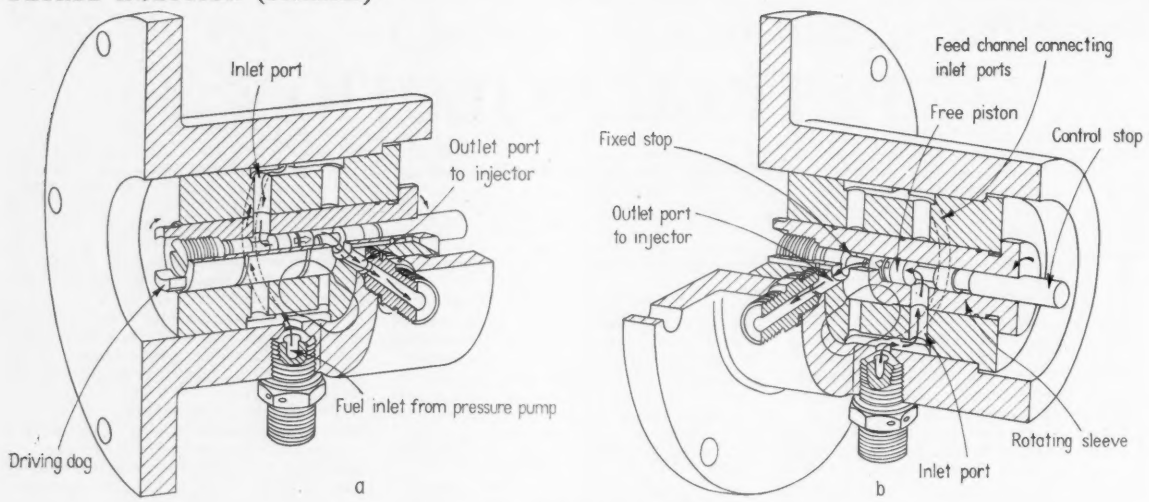


Fig. 1. Sectioned metering distributor

This discovery led to the development of a set of nozzles of the type shown, which gave an atomized spray from opening to closing, with no post-dribble, and which were later shown to be thoroughly reliable during 20,000 miles of service.

Space limitations necessitated the use of narrow spray angles of 40 deg., which meant some sacrifice in quality of atomization.

The Mixture Control. Fig. 3 shows the outline of the air manifold and the construction of a simple mixture control. During full throttle operation, air pressure is sensibly equal on both sides of the air piston, and the control tappet takes up a position determined by the initial loading on the springs A and B and the constant fuel pressure acting on the control stop. During part-throttle operation, the manifold depression is applied to the rear of the air piston and atmospheric pressure forces the piston and control lever to the left, reducing the shuttle travel, until a new position of equilibrium is found in which the tension spring C comes into use. Through this linear system of springs and levers, it is possible to obtain the characteristics of Fig. 4, which represent quite closely the fuel requirements of an engine. For full load operation, the mixture is made richer to obtain maximum power.

Bench Performance and Characteristics of System. After development, good and consistent results were obtained on rig tests from the completed system.

Fig. 5 shows the speed-delivery characteristics of the distributing unit for three positions of the control stop. Delivery is nearly independent of speed due to the fact that the shuttle is fuel-propelled at 100lb per sq. in, so that no filling troubles are experienced at high speeds. Delivery bears linear relation to stroke, a desirable feature when designing the mixture control device.

Table 1 shows the degree of uniformity obtained on the bench at either end of the speed range with various output settings, using "Pool" burning oil with standard nozzles and pipes. Oscillograph observations confirm that the individual injections from any one nozzle are, for practical purposes, identical.

TABLE 1. Uniformity of Delivery

Piston stroke, inches	Distributor, r.p.m.	To cylinder No. 1	To cylinder No. 2	To cylinder No. 3	To cylinder No. 4	Percentage variation
		Cubic millimetres per injection				
0·0062	250	8·6	8·6	8·8	8·8	+1·2 −1·2
	2,000	8·25	8·18	8·37	8·25	+1·3 −1·0
0·0125	250	17·25	17·0	17·25	17·25	+0·4 −1·1
	2,000	16·5	16·38	16·5	16·5	+0·2 −0·6
0·025	250	34·0	33·5	34·0	34·0	+0·4 −1·1
	2,000	32·75	32·5	32·5	33·0	+1·0 −0·6

The performance with the mixture control in action (Fig. 4) was settled after engine tests had established the relationship between manifold depression and fuel requirements.

Chief features of this system are:—

(1) The common bore and stroke (variable) used for metering, eliminates the problem of matching individual pumping elements.

(2) Metering is in no way dependent upon nozzle characteristics, and there is no need to match nozzle performance to close limits.

(3) The degree of atomization is constant over the whole of the speed- and load-range; the metering distributor has merely a metering and distributing function, and the nozzles in action are always at the constant line pressure.

(4) As nozzle line pressure is constant, accurate control over nozzle operation is possible.

(5) The start of injection is fixed relative to the engine. As the quantity of injected fuel is increased, the time of injection lengthens.

It is of interest to note that under idling conditions on a small engine, the quantity of fuel to be metered, distributed, and atomized for each cylinder is approximately one-third of a drop.

Engine tests

General Description. Engine work commenced on a four-cylinder 1,400 cu. cm engine with overhead valves and a compression ratio of 6·5/1. The metering distributor was mounted on the front chain-cover, being driven by a dog from an extension on the camshaft. The output of the fuel system was hand-controlled at the movable stop, and a special air manifold, well separated from the exhaust, had the usual butterfly throttle plate.

The aim of these tests was to enable a comparison to be made of the optimum performance of the engine with both petrol injection and carburettor systems. Data obtained in the course of these tests could then be used for the design of an automatic mixture control, and the associated ignition advance curves.

Exploratory Work—Positioning and Timing of Injectors. After preliminary running to confirm that the installed petrol injection system was functioning

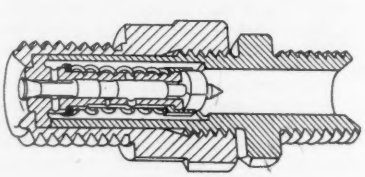


Fig. 2. Injector

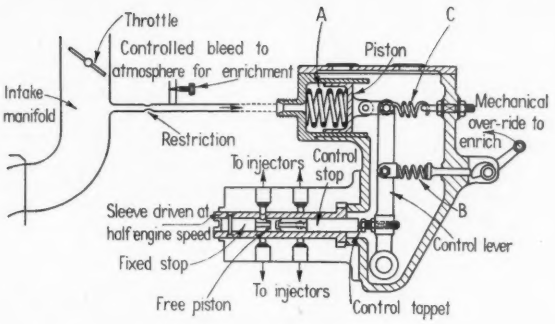


Fig. 3. Arrangement of metering distributor, mixture control, and manifold connections

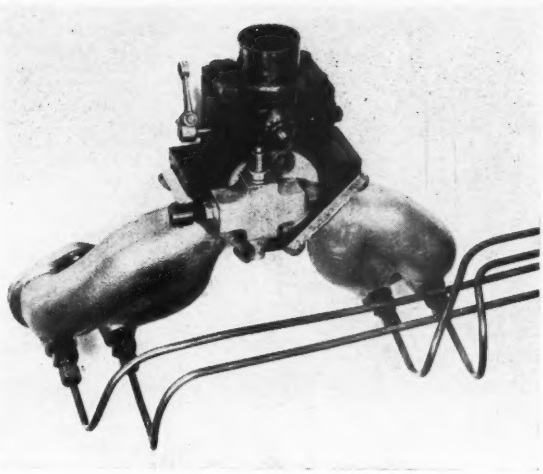


Fig. 6. Air manifold and injector location on 1,400 cu. cm. overhead valve engine

correctly, tests commenced to establish the effect on performance of varying the injector location and timing.

Cylinder head injection was tried first but gave disappointing results in economy. It appears that penetration of the spray is excessive for small cylinders, with the result that fuel is lost by contact with the cylinder walls, subsequently passing via the piston rings to the sump. This was confirmed by a sump oil-dilution check.

Inlet port injection gave better results except that some light-load missing occurred. Manifold injection at points about 2½ inches from the inlet valve cured this trouble and was found to give the best overall performance on this engine. Fig. 6 shows the positioning of the injectors and the general shape of the air manifold.

Timing was varied for all positions of the injectors, but did not appear to be at all critical. The final decision was to inject during the suction stroke, when the air velocity in that branch of the manifold was approaching a maximum. For bench test 45 deg. after top dead centre was used and later, on road tests, this was altered to 90 deg. after top dead centre.

An important advantage noted during these early tests was the smooth running of the engine, and the very stable idling. When coupled to the dynamometer, idling could be taken down to 150 r.p.m.; with a free engine the lowest speed was about 300 r.p.m.

Performance with Hand-controlled Mixture. Results derived from an examination of mixture loops taken over the speed range at full load for both carburettor and petrol injection systems with optimum ignition are shown in Fig. 7. This curve has been previously published (Brook and Nicolls 1948)* to illustrate the type of results obtainable with petrol injection.

Gain in power with petrol injection increased over the speed range up to a maximum of 8 per cent at 4,000 r.p.m. Specific consumption showed little difference above 2,500 r.p.m. but below this speed showed appreciable advantages, increasing to nearly 12 per cent at 1,000 r.p.m.

At part load it was evident that, with the petrol injection system, regular firing could be obtained with a much weaker average mixture strength than with the carburettor. The minimum specific consumption without missing was very little different for the two systems.

Performance with Mixture Control. Performance comparisons made by hand-varied quantities are of interest and define the target, but road results with either carburettor or petrol injection must depend upon the degree to which the automatic control fits the requirements of the engine.

The mixture control was next developed to give as nearly as possible the optimum performance as derived from the mixture loops, and final engine tests were run off to provide a direct comparison between petrol injection and the carburettor. The latter was a standard production unit with economy setting.

Figs. 8*a* and *b* show the results of this comparison. The petrol injection engine developed up to a maximum of 6 per cent more power at 3,250 r.p.m., but the full load specific consumption was similar to that of the carburettor over most of the speed range. The improvement over the carburettor in consumption at the part loads shown in Fig. 8*b* is between 18 and 26 per cent at 1,000 r.p.m. and between 10 and 15 per cent at 1,500 r.p.m. and above (secured by working at a weaker average mixture strength).

The full-load consumption at low speeds was disappointing, as the improvement available with hand control could not be realized. This is due to the difficulty in matching fuel- and air-flow through the engine and is illustrated in Fig. 9. Distributor fuel-delivery is not "speed-conscious" (Fig. 5) and is represented in Fig. 9 by a horizontal line, whereas cyclic air-consumption follows the volu-

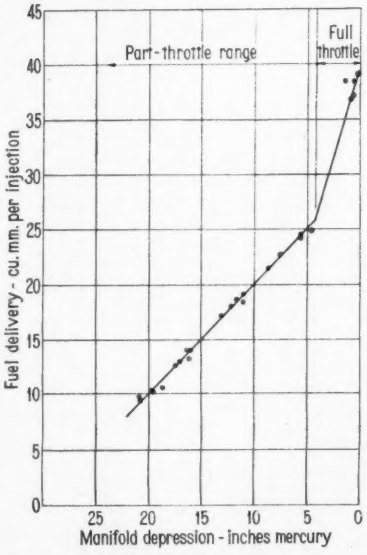


Fig. 4. Delivery characteristic of system with mixture control

Fuel pressure 100 lb per sq. in; "Pool" petrol; barometer 30·0 in mercury; inlet air temperature 19 deg. C.

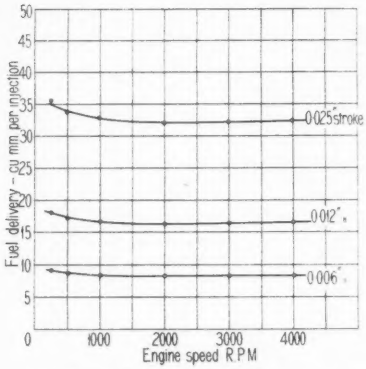


Fig. 5. Delivery characteristics of metering distributor

Fuel pressure 100 lb per sq in; standard nozzles; fuel, "Pool" burning oil

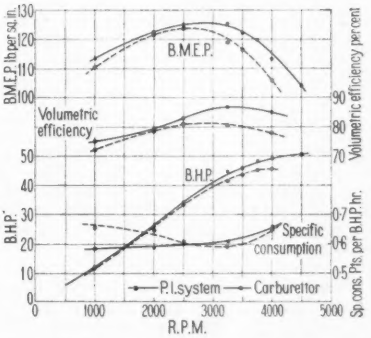


Fig. 7. Optimum full throttle performance of carburettor and petrol injection systems

1,400 cu. cm., four-cylinder overhead valve engine.

* An alphabetical list of references is given on p. 160.

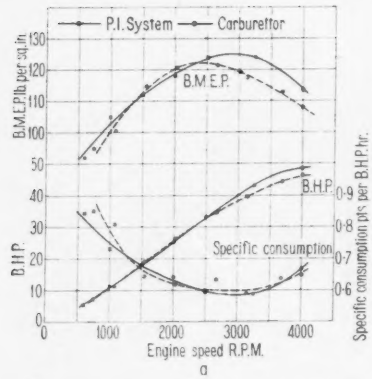


Fig. 8. Comparison between carburettor and petrol injection performance with mixture control

1,400 cu. cm., four-cylinder overhead valve engine; carburettor, economy setting and optimum ignition timing; petrol injection, with mixture control and automatic ignition timing.
a Full throttle performance.
b Part throttle economy.

metric efficiency curve of the engine. To obtain maximum power with economy at low speeds, the setting point shown in Fig. 9*a* should be chosen, but this would mean a real sacrifice in power at high speeds. Matching for maximum power in the mid-speed range in Fig. 9*b* has resulted in a mixture much richer than necessary at lower speeds, and in consequence no improvement in full load consumption over the carburetted engine has been possible.

The considerable improvement secured at part throttle has been obtained largely because of the inability of the design to burn a wide range of weak mixtures efficiently. The carburation system fails to distribute the mixture evenly, with the result that the carburettor setting has, in general, to be on the rich side to prevent cutting-out during acceleration. With the petrol injection system it has been possible to work at a weaker average mixture strength and thus secure a better specific consumption. It was left for road tests to establish, among other things, whether good pick-up and acceleration could be obtained from these lean settings without additional accelerating fuel.

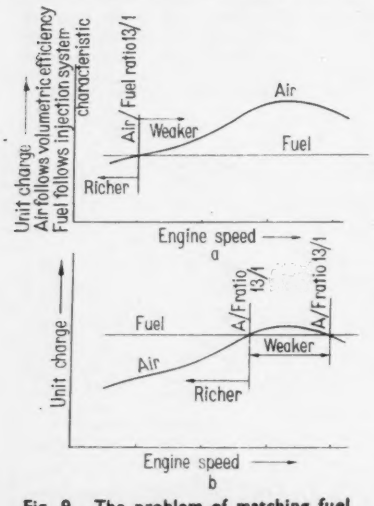


Fig. 9. The problem of matching fuel supply to air consumption under full load conditions

Road performance

Installation. Encouraged by the gains in power and economy on the bench, the engine was built into a four-door saloon-body chassis and replaced a 12 h.p. engine. The total weight, including the heavy and bulky source of pressure—now battery-driven and housed in the boot—was 28 cwt.

Installation difficulties necessitated the omission of the fan and the use of a copper gilled radiator, with about 20 per cent of the block cut away to make room for the metering distributor. The reduced cooling proved to be adequate for the engine throughout the year; in fact, after examination of the head at 6,400 miles, it was considered advisable to increase the thermostat setting from 70 to 85 deg. C.

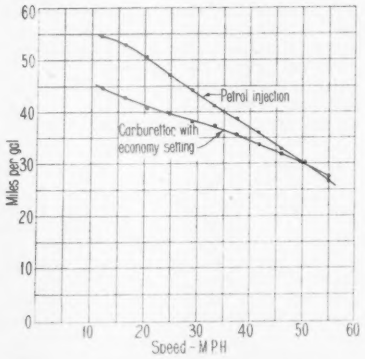


Fig. 10. Road miles per gallon at steady speeds

Four-door saloon body, laden weight, 1·54 tons; engine, 1,400 cu. cm., four-cylinder, overhead valve.

A Wilks flow-meter included in the suction line measured fuel input on the road in hundredths of a gallon and enabled consumption figures to be obtained under a variety of conditions.

After sufficient experience of performance and behaviour on the road had been gained (16,500 miles in eighteen months) the economy-set carburettor and its manifold were temporarily replaced in order to make a direct comparison.

Drivers' Impressions. Free-engine response was about the same as with the carburettor. Idling was very good at all temperatures. Road performance, as judged by maximum speed and ability to overtake, was much better, with a complete absence of "flatspots." There was noticeable equality of explosions, especially when ascending a steep hill in top gear at low speeds. The engine then fired uniformly and persisted at speeds as low as 10 m.p.h., whilst with the carburettor, power rapidly fell away and the engine stalled. The general impression was of a smoother and more powerful engine.

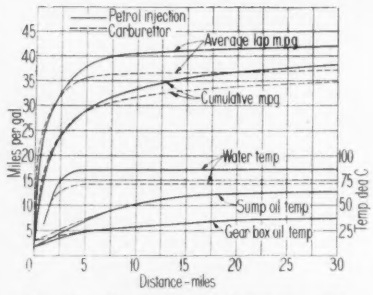


Fig. 11. Comparison between carburettor and petrol injection during warming-up run at 30 m.p.h.

Starting temperature: carburettor, 16 deg. C., petrol injection, 10 deg. C.

Startability. This was disappointing. The first trials were with the air-bleed enrichment device illustrated diagrammatically in Fig. 3, which allowed up to a maximum of full-throttle fuel for starting. In effect it is an air-bleed to the mixture-control cylinder with a restriction introduced at the manifold connection. Progressive closing of the bleed during drive-away reduces mixture strength to normal.

This was later found inadequate for winter use and a mechanical override was added which unloaded spring B in the control unit. For reasonable starting and drive-away at 20 deg. F., the available fuel had to be doubled, the approximate mixture strength then approaching carburettor practice at 3/1.

The technique adopted was to operate both devices for the start and drive-away, then cut the mechanical override, and finally reduce the bleed in stages until inoperative, final closure usually coinciding with a cylinder-head temperature of 50-60 deg. C.

Petrol Consumption. There are a number of practical difficulties involved in making accurate consumption records on the road due to the large number of possible variants. Road consumption figures are meaningless unless both the conditions are carefully stated and the quantities accurately measured; in particular, any quoted figure involving a start from cold can be quite misleading unless the temperature, nature of lubricants, and length of journey are stated.

Whenever possible, the comparative tests between petrol injection and the

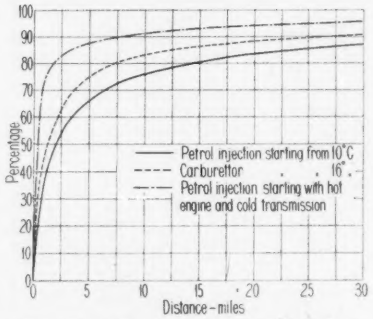

Petrol injection starting from 10° C
Carburettor " " 16° "
Petrol injection starting with hot engine and cold transmission
Percentage
Distance - miles

Fig. 12. Cumulative mileage per gallon as a percentage of potential mileage per gallon

Derived from Figs. 10 and 11.

carburettor have been made on the Motor Industry Research Proving Ground; wind and slight gradient effects have been countered either by use of a closed circuit (for the lower speeds); or averaging a number of runs in both directions (for the higher speeds). To reduce the effect of starting temperature, an S.A.E. 20W oil was used for the engine.

Where comparative figures are given under open road conditions they were obtained by the same driver.

Steady Speed Consumption. Fig. 10 shows the miles per gallon obtainable when hot under steady speed conditions in top gear with both petrol injection and the carburettor (economy setting). The gain of the petrol injection system is confined to the low-speed end of the range and progressively decreases up to 50 m.p.h. At 30 m.p.h. it is approximately 15 per cent; at 40 m.p.h. it is 6 per cent, at 50 m.p.h. they are equal.

Variable Speed Consumption. A comparison was made under forceful driving conditions. On a set course of 17 miles in the suburbs, involving built-up areas (30 m.p.h. limits to be observed), road junctions, halt signs, country lanes, and two stretches of three miles to be taken at 50–55 m.p.h., an overall course speed of 30 m.p.h. had to be obtained. Under these conditions, starting with the engine hot, the petrol injection car averaged 34 m.p.g. and the carburettor (economy setting) 30·3 m.p.g. (three runs each).

Long Journeys. For journeys over 50 miles the consumption of the petrol injection car varies somewhat with conditions but usually falls between 35 and 38 m.p.g. On a normal cross-country journey of 175 miles each way during the summer of 1949 the average speed was 35 m.p.h. and the overall consumption 37 m.p.g.

A comparison between petrol injection and the economy-set carburettor on long journeys, with the same driver, gave 38 m.p.g. for the former and 33 m.p.g. for the latter.

Short Journeys. The difference between long and short journeys lies in the greater influence on the overall consumption of the increased fuel, inefficiently used, for starting the engine and warming up the coolant and lubricants to the working temperatures.

The car was left standing overnight on the Motor Industry Research Association Proving Ground, and in the morning started up and driven at a constant 30 m.p.h. for one hour around the perimeter track while distances were recorded for every 0·01 gal. fuel used. From these figures both interval and cumulative m.p.g. were calculated and plotted. For purposes of reproduction, the interval m.p.g. have been reduced to average lap m.p.g. and plotted at the mileage at which half the fuel was used.

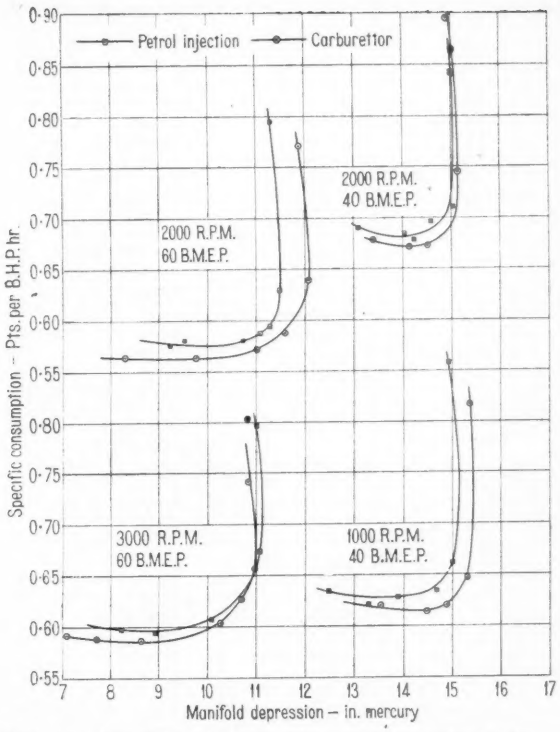


Fig. 14. Comparison between minimum consumption of carburettor and petrol injection systems in part-throttle range

Engine, 1,600 cu. cm., four-cylinder, compression ratio 7·35/1; optimum ignition timing; carburettor with variable main jet; petrol injection, downstream 30 deg. after top dead centre.

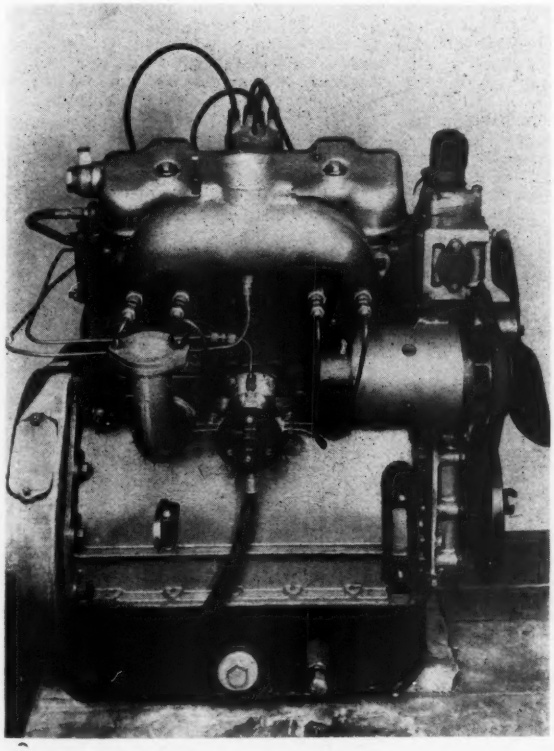


Fig. 13. Arrangement of petrol injection equipment on 1,600 cu. cm. four-cylinder engine

Fig. 11 shows the warming-up behaviour of the car with both petrol injection and carburettor systems under the above conditions. Initial engine temperatures were slightly in favour of the carburettor—16 as against 10 deg. C.

Consumptions in both cases are poor in the initial warming-up stages, and only begin to approach their maxima after about five miles' driving. From the temperature recordings it appears that the major influence on consumption is associated with head water temperature and that the subsequent small improvement after the thermostat takes control is due to the slower warming-up of the engine and transmission lubricants. In neither case does the lap average m.p.g. quite reach the figures obtained in the hot, steady speed, 30 m.p.h. condition (shown in Fig. 10), even after a 30-mile journey.

The poor consumptions at starting are reflected in the cumulative consumptions, and these show clearly that, on a twice-daily journey of five miles, the expectation for both systems is of the order of 29 m.p.g. At the end of a 30-mile journey the petrol injection car has an overall consumption of 38·5 m.p.g., which is 10 per cent better than the carburettor figure. There is some indication that the carburettor system, in spite of the initial handicap of having to replace at start the fuel evaporated from the float chamber during standing (0·01 gallon), does give better warming-up consumptions, but this may be due to the slight advantage in initial temperature noted above.

Fig. 12 shows the cumulative m.p.g. expressed as a percentage of the potential m.p.g. for each system, as obtained from the hot, steady-speed tests. In addition the corresponding results from a similar test with the petrol injection system, but starting with a pre-warmed engine (transmission cold) have been included.

Expressed in this manner, the curves show clearly the considerable effect of engine starting-temperatures upon short-journey consumptions. In this country, much present-day running is made up of short journeys, starting from cold, and it is of interest to note that a five-mile journey, with the carburettor system from a cold start of 16 deg. C., is likely to give only 75 per cent of the final hot consump-

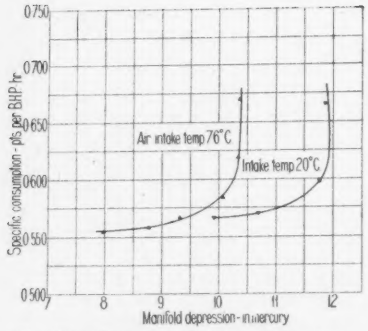


Fig. 15. Effect on specific consumption of adding heat to intake air

Engine: 1,600 cu. cm., four-cylinder, compression ratio 7.35/1; optimum ignition timing; mixture loops at constant power; 2,000 r.p.m.; 60lb. per sq. in. b.m.e.p.; petrol injection: downstream 30 deg. after top dead centre.

tion rate. It is certain that at lower temperatures, or with higher viscosity lubricants, the figures would have been much worse.

An apparent superiority of the carburettor system, as such (that is, with hot spot), is indicated from the percentage curves. It might well be that the petrol injection system could derive considerable benefit during the initial stages of warming-up by some interchange of exhaust heat to coolant.

Reliability. To date, 20,000 miles have been completed in eighteen months. The car has been used as private cars are used; home-to-work journeys, shopping, occasional long distance runs for business purposes. The only troubles encountered with the petrol injection system were associated with the lubrication of the engine-driven sleeve, which seized twice before adequate precautions were taken to ensure the exclusion of solid particles. No appreciable wear has been noted on any of the parts. The nozzles have been particularly trouble-free. No abnormal engine troubles have been experienced which could in any way be associated with petrol injection. The main difficulty has been the "source of pressure" which has remained an experimental improvisation of out-of-date equipment.

Application to a modern engine

Introduction. Recent work has been directed towards finding out whether any improvements could be secured on a modern engine capable of burning efficiently a wide range of lean mixtures. Development is not completed, but the results so far achieved enable direct performance comparisons to be made—at least under steady load conditions.

In its original form the engine has a compression ratio of 7·35/1 and has four cylinders totalling 1,600 cu cm capacity. It has overhead inlet and side exhaust valves and a hemispherical combustion chamber. The inlet and exhaust manifolds are on opposite sides of the engine, an arrangement which reduces inlet manifold heating from the exhaust and makes the engine particularly suitable for a petrol injection application. The salient features of this design have been described (Swaine, 1947).

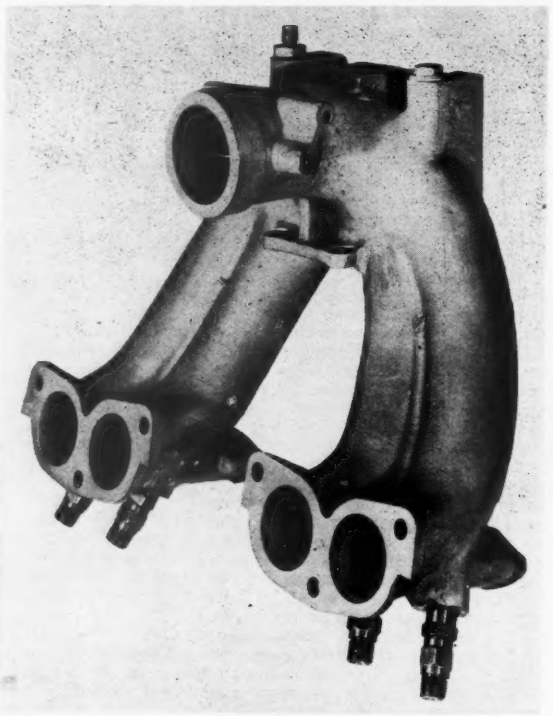


Fig. 17. Long branch air manifold used for both upstream and downstream injection

The petrol injection hand-controlled equipment used is the same as that already described, with minor modifications to such things as pipe lengths and nozzle holders, and the metering distributor is driven from a cross-shaft from the camshaft. The arrangement of the equipment is shown in Fig. 13. Manifold injection has been used throughout, and involved the trial of a number of manifolds to permit upstream and downstream injection, and to ease the final automatic control problem.

From among the many tests made with different manifolds and injector locations, results have been chosen which illustrate particular phases of the work. In this way, it is hoped to explain some of the principles governing the application of petrol injection to this class of engine.

Part Throttle Performance. As a result of a large number of tests, the opinion has been formed that the utilization of the fuel with the petrol injection system (at its present stage of development) is slightly inferior to the carburation system. The effect is noticeable in part-load consumption, which is about 2-3 per cent higher than the minimum obtainable with the carburettor.

Examples of mixture loops taken at constant torque with the two systems are shown in Fig. 14. An explanation of the difference in minimum consumption has been sought by endeavouring to improve the state of the mixture passing into the cylinder.

The results shown in Fig. 14 were obtained with downstream injection and nozzles located so that the unevaporated portion of the spray impinged upon the water-jacketed inlet port. The obvious ways of improving the mixing of fuel and air are:—

(*a*) To increase the relative velocities between fuel and air, that is, by upstream injection.

(*b*) To add heat to the incoming air.

(*c*) To supply finer atomization.

(*a*) Upstream injection gave results slightly better than downstream injection for both power and economy, and gas analysis confirmed that combustion was more complete. Minimum specific consumption was still just above that of the carburettor for part-load loops. There are, however, a number of practical disadvantages, the most important being the size limitation on diameter and length of the individual branches, which must be designed to avoid resonance. The pene-

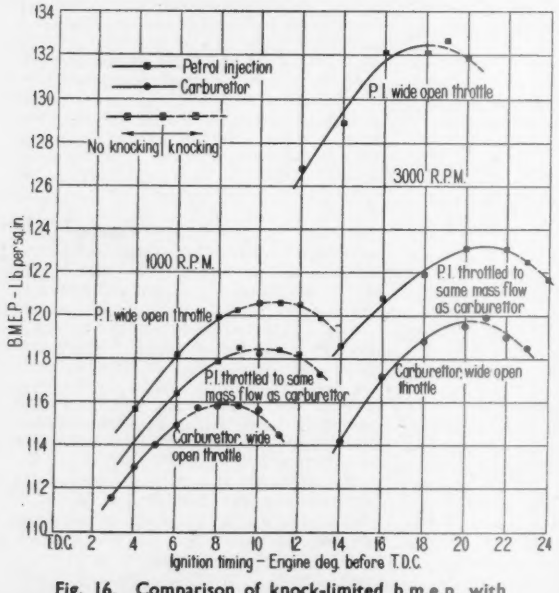


Fig. 16. Comparison of knock-limited b.m.e.p. with carburettor and petrol injection

Engine, 1,600 cu. cm., four-cylinder, compression ratio 7·7/1; fuel, unleaded spirit, 68 octane rating; air/fuel ratio 13/1; carburettor, standard manifold, variable main jet; petrol injection, upstream 55 deg. after top dead centre.

tration of the spray upstream at full load is negligible, but increases as the air density is reduced by throttling, so that lengthened branches (with a tendency to resonate) must be used to prevent fuel being carried to another cylinder. They must also be wide and straight, or deposition on walls results in a return to cold-surface carburation.

It was also found that when running at the weak limit of mixture strength, a "pop-back" involved all branches of the manifold instead of only one as in the case of down-stream injection. Another finding, as yet unexplained, is the increased tendency to knocking at part loads with upstream injection.

(*b*) The addition of heat to the intake air was expected to assist in the evaporation of the fuel and to lower the pumping losses for a given power output.

A comparison between loops taken under identical conditions of speed and load, with the single exception of inlet air temperatures (20 and 76 deg. C.) is shown in Fig. 15 and indicates a small improvement in brake specific consumption using the heated air intake. The decrease in pumping losses is chiefly responsible for this small difference.

As further confirmation that added heat is not materially assisting the vaporization of the fuel, a test was made by running with progressive addition of heat to the air intake, but maintaining a constant mass air-flow and fuel supply. In this case, the indicated specific consumption did not vary by more than 1 per cent over the inlet temperature range of 21–91 deg. C., showing no worthwhile advantage in vaporization due to heated air.

Light knocking was just setting in with the air temperature at 91 deg. C. and the air-heating necessary to have appreciable effect on brake thermal efficiency in this range of half-load running will certainly introduce the problem of detonation.

(*c*) Finer atomization is a field which has not yet been thoroughly investigated. Efforts have been made to maintain the system pressure at 100lb per sq in for reasons of economy and pump life, with the result that finer atomization can then only be obtained by nozzles of smaller orifice and consequent risk of stoppages. Preliminary tests with nozzles of about 60 per cent of the flow rate at 100lb per sq in have shown no improvement.,

Knock-limited b.m.e.p. Preliminary tests with a petrol injection system are almost certain to produce more power, due to the obvious improvement in volumetric efficiency. There is then a great temptation to be satisfied with the results, provided that the improvement obtained agrees with the increase in air consumption.

However, the use of a petrol injection system which is capable of accurate metering in conjunction with a satisfactory air manifold, will produce further increases in power, for :—

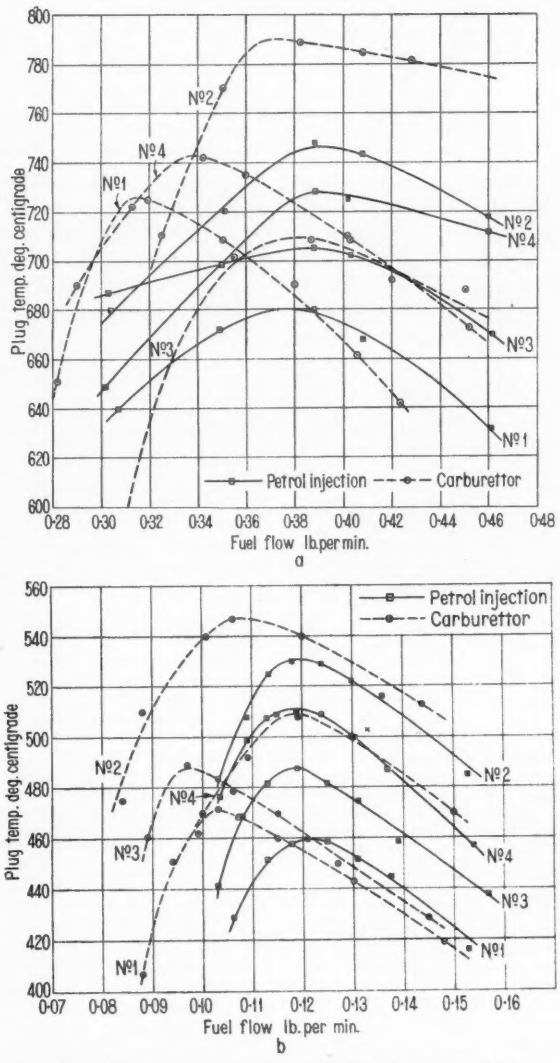


Fig. 18. Plug temperature with carburettor and petrol injection

Engine, 1,600 cu. cm., four-cylinder, compression ratio 7·35/1; carburettor, standard manifold, variable main jet; petrol injection, long branch manifold downstream 30 deg. after top dead centre.
a Full throttle, 3,000 r.p.m.
b Full throttle, 1,000 r.p.m.

(*a*) The equality of air, fuel, and anti-detonant results in all cylinders requiring the same ignition-timing, and in consequence the performance of the engine is not limited by the ignition timing of any one cylinder as in the case of maldistribution.

(*b*) A cooler charge and combustion chamber permits earlier ignition timing without detonation. Alternatively the compression-ratio can be slightly increased.

Fig. 16 shows how the improvement obtainable with petrol injection is made up at both 1,000 r.p.m. and 3,000 r.p.m. The curves have been obtained from tests made with upstream injection into a manifold with long individual branches to each cylinder. This manifold, also used for downstream injection, is illustrated in Fig. 17. An unleaded reference-fuel, of 68 octane rating, was used and the compression-ratio was 7·7/1. Air/fuel ratio was maintained constant at 13/1 by measurement of both air and fuel.

B.m.e.p. has been plotted against ignition advance for:
(*a*) The carburettor with wide-open throttle.
(*b*) Petrol injection controlled to give the same mass air- and fuel-flow as the carburettor in (*a*).
(*c*) Petrol injection with wide-open throttle.

When the same mass air/fuel flow as for the carburettor is delivered to the engine through the petrol injection system, there is an increase of 3 per cent in knock-limited power at both speeds. It should be noted that this also represents a 3 per cent decrease in specific consumption.

With a non-detonating fuel, the increase is just over 2 per cent at 1,000 r.p.m. and just below 3 per cent at 3,000 r.p.m.

With wide-open throttle (*c*) a further increase of 1½ per cent at 1,000 r.p.m. and 7½ per cent at 3,000 r.p.m. is obtained as a result of better filling.

This indicates that correct ignition-timing, mixture distribution, and a lower charge-temperature are significant contributory factors in power increase at low speeds, where only a small increase in volumetric efficiency is to be expected.

At 3,000 r.p.m., intake resonance resulting from the use of a long-branch manifold is responsible for about one-third of the increase noted.

Cylinder-head Temperatures. Maximum temperature in the cylinder is one of the factors limiting power output for a given design. Tests made with thermocouple sparking plugs have shown that the temperatures reached with the carburettor system are considerably higher than with either upstream or downstream petrol injection. Fig. 18 shows the temperatures recorded during tests for a mixture-distribution survey at 1,000 r.p.m. and 3,000 r.p.m., with both carburettor and down-stream petrol injection. At 3,000r.p.m. the petrol injection engine was developing a maximum of about 11 per cent more power, although the plug temperature of the hottest cylinder (No. 2) was 45 deg. C. lower than with the carburettor. Tests indicate that plug temperatures with upstream injection are intermediate between those obtained with down stream injection and the carburettor.

Elimination of manifold heating is responsible for a small reduction in temperature, the remainder being due to fuel evaporation in the head in the case of downstream injection. With upstream injection, the heat for evaporation must be taken largely from the air surrounding the manifold.

It will be noted that the temperature peaks in Fig. 18 for the petrol injection system occur at approximately the same fuel flow, indicating mixture distribution very close to equality.

Advantage was taken of the lower cylinder temperatures to increase slightly the compression ratio from 7·35 to 7·7/1.

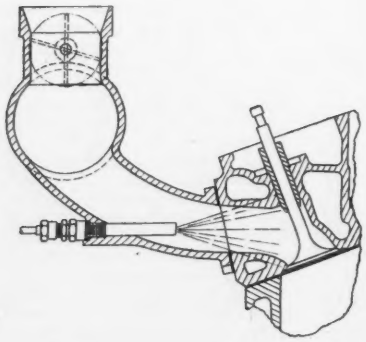


Fig. 19. Air manifold and nozzle location for downstream injection

Air Distribution and Manifold Design. A petrol injection system in which the fuel is accurately metered must obviously work in conjunction with an air manifold which distributes equal cyclic air-charges to all cylinders. In order to assist the control problem of this system, the manifold was also required, in conjunction with the valve-timing of the engine, to give a reasonably flat volumetric efficiency curve.

Checks with oscillograph recordings of the compression pressure over a large number of cycles showed that cyclic variation was not a problem, being less than 1 per cent.

A target figure of ± 2 per cent was set for air distribution and was achieved without difficulty. A direct method was used to check this, by measurement of the air exhausted from each cylinder under motoring conditions. This method also confirmed how resonance in the intake branches altered the shape of the volumetric efficiency curve, as had been found when working on the engine with the long-branch manifold shown in Fig. 17. This manifold caused a drop in the power output at 2,000 r.p.m., where it was below that of the carburettor, while at 3,000 r.p.m. the power was boosted as shown in Fig. 16.

Finally a manifold, suitable for downstream injection, shown in Figs. 13 and 19, was evolved, consisting of four tapering branches opening from a large common reservoir of about 50 cu. in. capacity. This reservoir reduces pressure-wave interaction between branches, and the manifold has made little alteration to the basic volumetric efficiency curve of the engine.

Power Curves. Representative full-load performances from this new manifold are compared with the carburettor performance in Fig. 20.

The compression ratio of the engine is 7·7/1 and optimum knock-free timing is used for both systems. A reference fuel (68 octane) was used for tests at this compression ratio and petrol injection fuel delivery was controlled at a fixed setting, which will eventually be provided by the mixture control. The results with a slightly richer alternative petrol injection setting are also shown. The carburettor had the standard jet settings.

Ignition has to be retarded below 2,500 r.p.m. with the carburettor at this compression ratio with 68 octane fuel, and a better performance below 2,500 r.p.m. is obtained at the original compression ratio of 7·35/1.

The excellent petrol injection full-load consumption should be well reflected in fuel saving under traffic driving conditions and is secured by working towards the base of the mixture loops, that is, about 3–6 per cent below maximum power.

At 4,000 r.p.m., power output is 17 per cent above the carburetted engine with specific consumption 15 per cent better, while, with the slightly richer alternative setting shown, there is 19 per cent more power at 12 per cent lower specific consumption.

Conclusions

The results described have been obtained on two high-speed engines, of approximately 1½ litres capacity, but of widely differing characteristics, with a system which has metered fuel to the cylinders within a tolerance of ± 2 per cent. Although it is evident that further improvement would arise from development aimed at better fuel-air mixing, the application of petrol injection has in each instance produced an appreciable improvement in performance. The results of the tests may be summed up as follows :—

(1) Petrol injection is unlikely to give any considerable improvement over the part-throttle range if the engine is capable of burning efficiently a wide range of weak mixtures.

(2) Petrol injection may give part-throttle improvement on an engine not specifically designed to run with very weak mixtures. In such a case, by operating at weaker average mixture strengths (possible because of better distribution), economy has been improved by 10-15 per cent.

(3) Petrol injection will give increased power with increased economy due to :—

(*a*) Better filling, resulting from removal of the choke, redesign of the manifold and absence of intake charge heating.

(*b*) Reduced octane requirement, resulting from the cooler and equalized charge of fuel and anti-detonant.

(*c*) A cooler combustion chamber, permitting the use of weaker mixtures at full load without overheating.

(*d*) The use of a higher compression ratio, alternative or in conjunction to (*b*) and (*c*).

An increase of power amounting to 19 per cent has been obtained on one engine with an increase in specific economy of 12 per cent.

(4) Petrol injection on the road, apart from the advantages resulting from increased power, is pleasanter to drive and is characterized by engine smoothness resulting from the equality of torque impulses, persistence of effort at full throttle on gradients where the carburetted engine lost power and stalled, and better idling.

(5) Petrol injection startability at low temperature is not appreciably better and requires about the same amount of over-fuelling as the carburettor. It should, however, be possible to secure better control of the fuel during the warming-up process, when the manifold is cold and the distribution difficulties of the carburettor system are accentuated.

(6) Steady speed, hot consumption-tests on the road with the two systems confirmed the part-throttle economy obtained on the bench with the first engine, but brought out the need for careful definition of the conditions when quoting fuel consumption figures. The influence of the fuel used inefficiently during starting and warming-up has a considerable effect upon overall short-journey consumption figures. There is some indication that the carburation system is more economical in warming-up, in spite of having to replace fuel evaporated while standing.

Throughout this work no alteration has been made to the basic engine design, and in both cases the petrol injection system

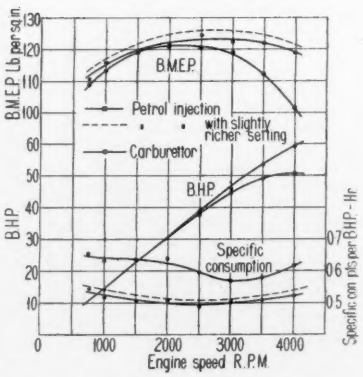


Fig. 20. Comparison between carburettor and petrol injection full throttle performance

Engine : 1,600 cu. cm., four-cylinder, compression ratio 7·7/1 ; fuel : unleaded spirit, 68 octane rating ; optimum knock-free ignition ; carburettor : standard setting ; petrol injection : 55 deg. after top dead centre, downstream manifold as in Fig. 19 ; fixed fuel setting.

can be said to have been carefully grafted on to an existing engine. The substitution of a petrol injection system for carburation does give a measure of freedom in design once manifolding is divorced from the necessity of carrying fuel, and it is evident that the characteristic torque/speed curve can then be modified. High torque at low speeds can be obtained by a compromise in valve timing which need not involve too great a sacrifice in the high-speed range.

There appears to be no reason why warming-up should not also be improved by suitable use of waste heat during the early stages, a development which would require some modification to existing exhaust and coolant layouts.

While this development has been undertaken on four-cylinder engines, the advantages enumerated should apply with equal, if not greater, emphasis to six-cylinder engines.

Finally, it must remain for each individual manufacturer or user to decide whether the many advantages enumerated make petrol injection a worth-while proposition for his particular needs.

Acknowledgements. The authors wish to acknowledge the encouragement and help which they have received from the Rover Company, Ltd., and to thank the directors of Joseph Lucas, Ltd., for permission to publish these results.

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BROOK, K., NICOLLS, W. E. W. 1948 I.Mech.E. Automobile Division Proc. 1947-48, p. 30, "Some Aspects of Petrol Injection Equipment Development".

SWAINE, J. 1947 I.Mech.E. Automobile Division Proc. 1947-48, p. 51, "Some Design Aspects of Poppet-Valve Cylinder Heads, for Spark-Ignition, Liquid-Cooled Engines".

Specialloid Ltd., North Finchley, London, N.12, have issued a brochure dealing with single piece core, hydraulically die cast pistons for diesel engines. Detail illustrations are included showing a range of the more popular diesel engine pistons manufactured in substantial quantities, and there are also notes on the design features, material and production methods.

THREAD GRINDING

New Centreless Methods for Headed Pieces

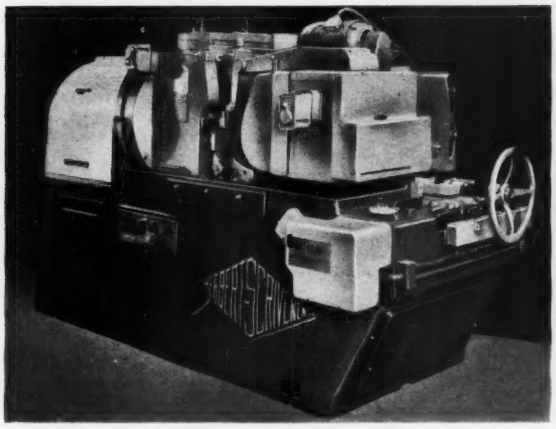


Fig. 1. Scrivener centreless thread grinder for straight through or headed work.

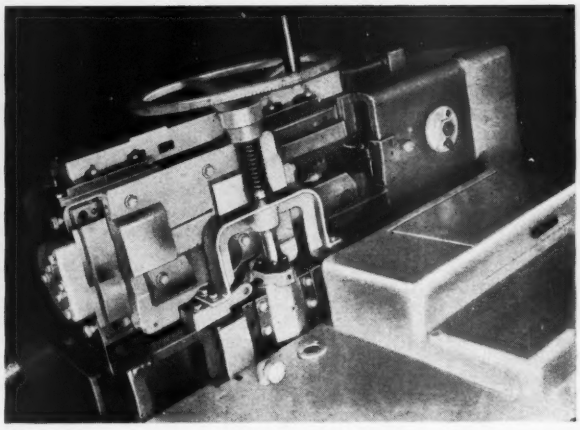


Fig. 2. The controlled-cycle slide on the Scrivener universal thread grinder.

ALTHOUGH the centreless thread grinding process is applicable to plunge-grinding headed pieces such as bolts as well as to straight-through grinding of components without heads, there are important differences in procedure. Plunge grinding for producing threads on headed work may be divided into two classes: (1) grinding the threads from the solid, and (2) correcting threads that have been rough machined. At the present stage of development it is doubtful whether the centreless thread grinding process will displace the usual methods of forming threads from the solid, except in stainless and other special steels that are difficult to machine. For such materials grinding is a commercial proposition, despite the fact that for a 1in by 8 t.p.i. Whitworth thread ground from the solid four or five cuts may be necessary to produce the desired standard of finish.

There is no doubt that up to the present, for headed work centreless thread grinding has been most widely used for correcting and finishing preformed threads. This is not surprising. Recent years have seen great developments in the use of high-tensile bolts of all kinds and at the same time there is a demand for more exacting standards of accuracy and finish in place of the old "commercial" limits. Arthur Scrivener, Ltd., Tyburn Road, Birmingham, who are pioneers of the centreless thread grinding process, have recently developed improved mechanism for dealing with headed pieces. The machines are produced by the company under Herbert-Scrivener patents.

It will be appreciated that in centreless grinding headed components, the cutting action of the grinding wheel must be limited to little more than one revolution of the work. While conforming to this fundamental, the technique with the Scrivener universal machine shown in Figs. 1 and 2 presents several interesting divergences from normal practice. The most marked divergence is that the work is not rotated until the grinding wheel has penetrated to the full depth of cut. For plunge grinding, the machine embodies the well-known controlled-cycle principle for advancing the work against the wheel. Furthermore, the cam plate employed in addition to its mechanical function of advancing the control-wheel head and the work as in plain centreless grinding, also actuates various circuits during the advance and return stroke. Among the functions it actuates are turning the coolant supply on and off and starting and stopping rotation of the control wheel.

The complete cycle for grinding or correcting a headed workpiece on the controlled-cycle machine is:—

(1) The component is placed on the workplate and located by means of an overhead finger. A lever is then depressed and the fully automatic cycle is started. Depression of the lever sets the cam plate in motion to advance the control wheel, which is not rotating, and the work up against the formed grinding wheel.

(2) As the control wheel and the work advance a trip turns on the coolant.

(3) When the advance of the cam plate has forced the non-rotating work on to the grinding wheel to the full depth of cut, a second trip momentarily stops the movement of the cam plate and at the same time starts the slow rotation of the control wheel to cause the work to rotate.

(4) When the work has completed 1½ revolutions, in other words, has advanced between the wheels a distance equal to 1½ pitches of the thread, contact is made with a stop. This energises a solenoid to change-over a hydraulic valve and raise the actuating lever. As the lever is raised the return movement of the cam plate starts and the wheels begin to open.

(5) As the wheels open a trip stops the rotation of the control wheel and another trip shuts off the coolant.

(6) When the wheels are fully open another trip de-energises the solenoid so that the cycle can be repeated.

As with ordinary centreless grinding, the plunge-grinding of threads on headed workpieces is a longer operation than straight-through thread grinding on work such as socket screws that can be passed straight through between the wheels. In the case of a 1in × 8 t.p.i. Whitworth thread with 0.020in left on for correction, centreless grinding would be carried out in two cuts. At the first 0.15in and at the second 0.005in would be removed. The complete grinding cycle would take some 45 seconds.

JOINT ENGINEERING CONFERENCE, 1951

A JOINT Engineering Conference from June 4th to 15th, 1951, is to be arranged by the Councils of the Institutions of Civil, Mechanical and Electrical Engineers. It will coincide with the Festival of Britain, and the theme of the Conference will be to place on record the achievements of British engineers and it will also afford an opportunity for engineers from all parts of the world to discuss the future trends of development. (1875)

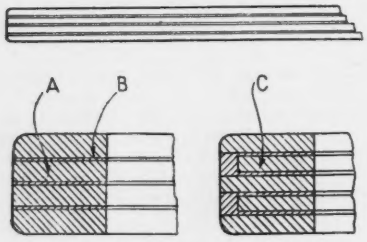
CURRENT PATENTS

A Comprehensive Review of Recent Automobile Specifications

Anti-scuffing piston ring

THE advantage of a soft metal coating for the initial running-in period of a ring is by this construction continued throughout the entire working life. Each ring is built up of a series of cast-iron or steel laminæ A sweated together by thin layers B of a softer metal, such as tin, cadmium, zinc or a suitable solder. The soft metal must have a melting point below the temperature at which the temper of the steel ring would be impaired (900-1,100 deg F) and above the working temperature (500-600 deg F).

These composite rings are produced by coating, dipping or plating the steel laminæ, assembling in a pile, consolidating under heat and pressure, and machining to finished size. Proportions are not critical and a typical ring for a 0.125in groove could have four laminæ of 0.0295in thickness spaced by soft metal layers of approximately 0.003in.

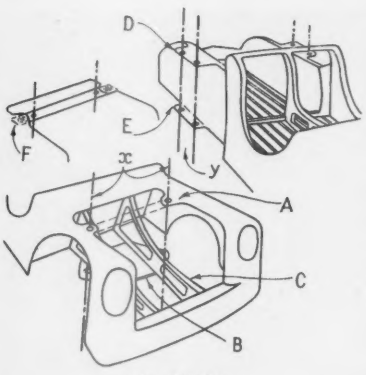


No. 614753.

In a modified construction to provide additional soft metal for the breaking-in period, intermediate laminæ C may be of reduced outer diameter. *Patent No.* 614753. *A. Abbey (Ramsey Corp., U.S.A.).*

Wheel mounting for shell-type bodies

THIS vehicle is of the axle-less type and has a shell-type body of relatively thin sheet metal. As it is unsatisfactory to attach wheel mounting devices directly to the sheet metal body, this construction uses vertical spindles carried from integral parts of the body or from integral parts of sheet metal members secured to the body.



No. 613518.

For the front end of such a vehicle, vertical mounting spindles on axes *x* are carried from extensions A on the metal body and the free ends of cross-piece B secured below the longitudinal bearer members C which receive the engine, steering gear and other components. Spindles serve a duel function of furnishing a mounting for the wheel stub axles and a strengthening element for the body. In specific applications such spindles may also be used as guides during vertical or rotational movement of the stub axle.

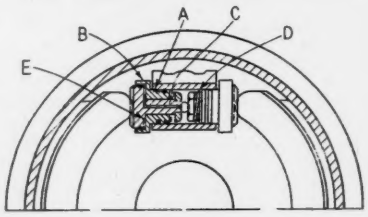
At the rear, two spindles on axes *y* may be provided for each wheel. These are secured to a flange D bent from the upper run of the body side wall and an extension E of the floor member of the body. If desired the structure may be braced by sheet metal cross-pieces F located at upper and lower points of spindle attachment and spaced from the body by rubber damping blocks. *Patent No.* 613518. *S. A. Des Usines Chausson (France).*

Automatic compensation of brake shoe clearance

IN a hydraulically actuated brake, of the type having only a very small clearance between the friction surfaces when inoperative, automatic adjustment of clearance to compensate for wear is furnished by duplex plungers in the brake cylinder. Plunger A is formed with a cylindrical chamber B and an axial bore in which is slidably arranged a two-diameter plunger C. A drilled passageway in plunger C brings chamber B into open fluid communication with the brake cylinder. Sealing is effected by a single ring on plunger C and a pair of rings, offering greater frictional resistance to movement, on plunger A.

When at rest the inner plunger is retracted by spring D; the movement being limited by a stop formed by shoulder E on the plunger. On application of fluid pressure, plunger C moves out to take up the clearance and apply the shoe to the drum. Pressure in chamber B prevents plunger A from following unless the clearance is sufficient to take up the permitted relative movement. In that case the frictional resistance of plunger A is overcome and it is moved out to a new adjusted position.

In an alternative construction a resilient diaphragm attached to the end of a cylindrical inner plunger seals the chamber B. *Patent No.* 614845. *J. Lucas, Ltd., and R. J. Ifield.*



No. 614845.

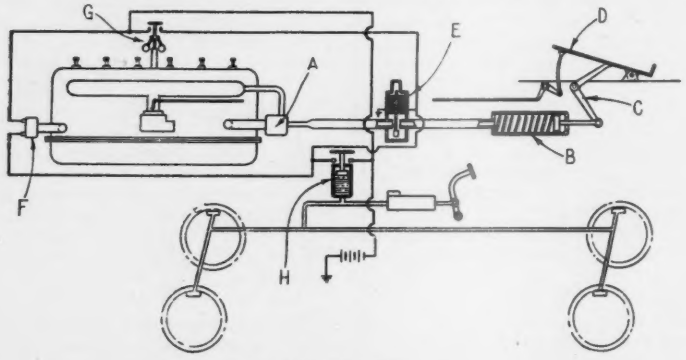
Control warning and safety system

THE object of this invention is to provide a control mechanism which prevents or makes difficult the operation of the accelerator pedal when any one of a variety of engine or vehicle conditions are abnormal. Either the manifold depression or the lubricating oil pressure may be used to operate the system.

The diagram shows a thermostatic valve A, closed when cold, located in a pipe connecting the engine induction manifold to a cylinder B in which a piston is constrained by a spring to move bell-crank lever C and oppose depression of the accelerator pedal D. In the pipe-line is also an electromagnetic valve E which can interrupt pasage through the pipe. When both these valves are open, engine suction acts on the piston and withdraws lever C from contact with the accelerator.

A thermally responsive switch F closes under excessive temperature of the coolant to actuate valve E, and a centrifugally operated switch G has the same effect if the engine is over-speeded. A check on the condition of the hydraulic braking system is provided by a switch H operated by a float in the fluid reservoir.

Further switches may be provided in the circuit of valve E to give warning of insufficient or excessive oil pressure or of improperly closed doors. *Patent No.* 613936. *M. Dach (U.S.A.).*



No. 613936.